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DEVELOPMENT OF A LIQUID SHROUD CRYOGENIC SUPERCRITICAL PRESSURE STORAGE SYSTEM

Prepared by

THE BENDIX CORPORATION

Davenport, Iowa

for Manned Spacecraft Center

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION • WASHINGTON, D. C. • MAY 1968

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DEVELOPMENT OF A LIQUID SHROUD CRYOGENIC SUPERCRITICAL PRESSURE STORAGE SYSTEM

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NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

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FOREWORD

This report has been prepared by the Bendix Corporation's Instruments & Life Support Division, Davenport, Iowa, under Contract NAS 9-4634. The work described in the report was conducted under the sponsorhip of the Propulsion and Power Division of the NASA Manned Spacecraft Center, Houston, Texas. The technical representative was Mr. Gordon Rysavy, EP5, Power Generation Branch, Power and Propulsion Division of the NASA Manned Spacecraft Center, Houston, Texas. Mr. Robert Lundeen was the project leader for the Bendix Corporation. Dr. George H. Bancroft, Dr. Blase J. Sollami, Mr. Dale Hankins and Mr. James A. Mientus provided technical consultation, while Chief Cryogenics Engineer Paul J. Gardner provided administrative supervision. The report summarizes work begun 2 June 1965 and concluded 16 November 1966.

ABSTRACT

The concept of insulating a cryogenic fluid by the vaporization of a secondary cryogenic refrigerant is not new. This technique has been used for several years in ground equipment and laboratory type liquid helium dewars. However, little effort has previously been expended in applying this technique to flight-type equipment. Fuel cell reactants supply systems, environmental control systems, and propellant pressurization systems to be utilized aboard future spacecraft will all require improved state-of-the-art advances in the handling and storage of cryogenic fluids. The concept of shroud cooling of a cryogenic storage system is discussed in this report to determine the potential weight savings and improved thermal performance for certain applications. In addition, vapor cooling of a discrete radiation shield for more effective insulation is discussed. The Cryogenic Shroud System fabricated and tested in this program is described. The feasibility of this cryogenic storage system for specific flight applications has been successfully demonstrated by this program.

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LIST OF SYMBOLS

h =	Specific enthalpy of fluid at given thermodynamic condition	Btu/Lb
$\dot{m}$ =	Mass flow rate	Lb/Hr
Q =	Rate of heat transfer into system	Btu/Hr-Ft^2
Q' =	Volumetric refrigeration capacity of shroud fluid	Btu/Ft^3
T_1 =	Absolute temperature of enclosed surface in a discrete shielded dewar	^oR
T_2 =	Absolute temperature of enclosing surface in a discrete shielded dewar	^oR
σ =	Stefan - Boltzmann constant	Btu/Hr Sq/Ft$^oR^4$
E =	Emissivity factor between surfaces in a discrete shielded dewar; a function of the emissivities of the surfaces	
A =	Area of enclosed surface in a discrete shielded dewar	Ft^2
n =	Number of discrete shields	
V_s =	Usable shroud volume	Ft^3
A_s =	Outer surface area of shroud vessel	Ft^2
A_{si} =	Outer surface area of laminar insulation	Ft^2
θ =	Standby time	Hr.
t =	Vessel wall thickness	In.
P =	Operation pressure	Lb/Sq. In.Abs.
D =	Inside vessel diameter	In.
UTS =	Ultimate tensile strength of material	Lb/Sq. In.
Y =	Modulus of elasticity	Lb/Sq. In.
γ =	Poisson's ratio	
SF =	Safety factor	

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I INTRODUCTION

Since anticipated future space flights are to be of long durations, this will necessarily create greater demands on the environmental system supplies. It is therefore a prime requisite that a method be developed to preserve these supplies during standby conditions. As a result, the Bendix Corporation has designed and fabricated a cryogenic storage system which employs the use of a shroud unit to fulfill this function.

This shroud concept provides a means of using a secondary fluid to act as a refrigerant and thus greatly reduce loss of the primary fluid contained in the inner vessel. Two separate shroud designs were investigated during this period of development and are thoroughly elaborated upon within the report. The integrally-mounted shroud system has the secondary fluid surrounding and in contact with the outer surface of the inner vessel, while the isothermally-mounted shroud system has the secondary fluid contained within the shroud itself.

In addition to exploring the shroud concept, this program further explored the use of a vapor-cooled discrete radiation shield as a means of reducing heat loss of the stored fluids. The use of a vapor-cooled discrete radiation shield proved highly effective when used with the shrouded cryogenic storage design in that it maximizes the utilization of the vented secondary shroud fluid.

The shroud system was shown to be an effective design for both pre-launch and in-flight standbys. It is especially effective for reducing losses during pre-launch standby since refilling of the shroud unit can be done as often as necessary. It was also found that a hydrogen-shrouded helium storage system represents an improved method of storing helium at high densities as well as eliminating loading problems currently experienced with conventional helium storage systems.

II REVIEW OF SHROUD CONCEPT

The major benefit realized from utilizing an expendable secondary cryogenic fluid to refrigerate another cryogen in a flight storage system is the extension of the system standby time. This would encompass the period of time between loading the primary fluid in its storage vessel and its initial removal for either pressure relief or supply usage.

The refrigerant may be required only for extension of pre-launch standby, after which the secondary fluid will be expelled, or will have been completely vaporized. Or, for extended mission durations, wherein post-launch standby is extremely long, the secondary fluid can be carried on board and reserved for in-flight cooling both prior to and following the initial withdrawal.

The vaporization cooling capacities that are available from several cryogenic liquids are shown in Figures 1 and 2. These figures represent for each fluid the latent heat of vaporization as well as the quantity of heat required to increase the temperature of the fluid to -100°F. This temperature was arbitrarily selected to provide a reasonable comparison of the cooling capacities of the liquids when vaporizing within an insulated dewar. The actual temperature of the vaporized liquid exiting from a dewar into the atmosphere depends upon the fluid used, its initial thermodynamic state, and the thermal insulating characteristics of the dewar.

Figure 1 shows that hydrogen has the greatest cooling capacity per unit mass, followed by the other fluids in order of increasing density, with the exception of neon and oxygen. Based upon volume, the high density fluids are more efficient, with oxygen being the most effective. Figures 1 and 2 both show that there is considerable cooling available for most of the fluids.

To utilize the cooling capabilities of these cryogenic fluids, it is necessary to select a storage system design that will result in the efficient interception of heat by the secondary refrigerant. Thus, by making it possible for the secondary fluid to intercept this heat, most of the heat transfer to the primary stored fluid is prevented.

There are presently two designs applicable to flight storage systems which are based on the use of a secondary fluid as a refrigerant. These two systems are the isothermally-mounted shroud system shown schematically in Figure 3 and the integrally-mounted shroud system shown in Figure 4.

The design of the two systems is similar in that the secondary refrigerant vessel surrounds the primary storage vessel. However, the isothermal design does not allow the shroud fluid to physically contact the primary vessel, as does the fluid in the integral system.

ISOTHERMALLY-MOUNTED SHROUD DESIGN

The isothermally-mounted shroud system consists of a shroud vessel completely surrounding a spherical inner storage vessel, with both vessels

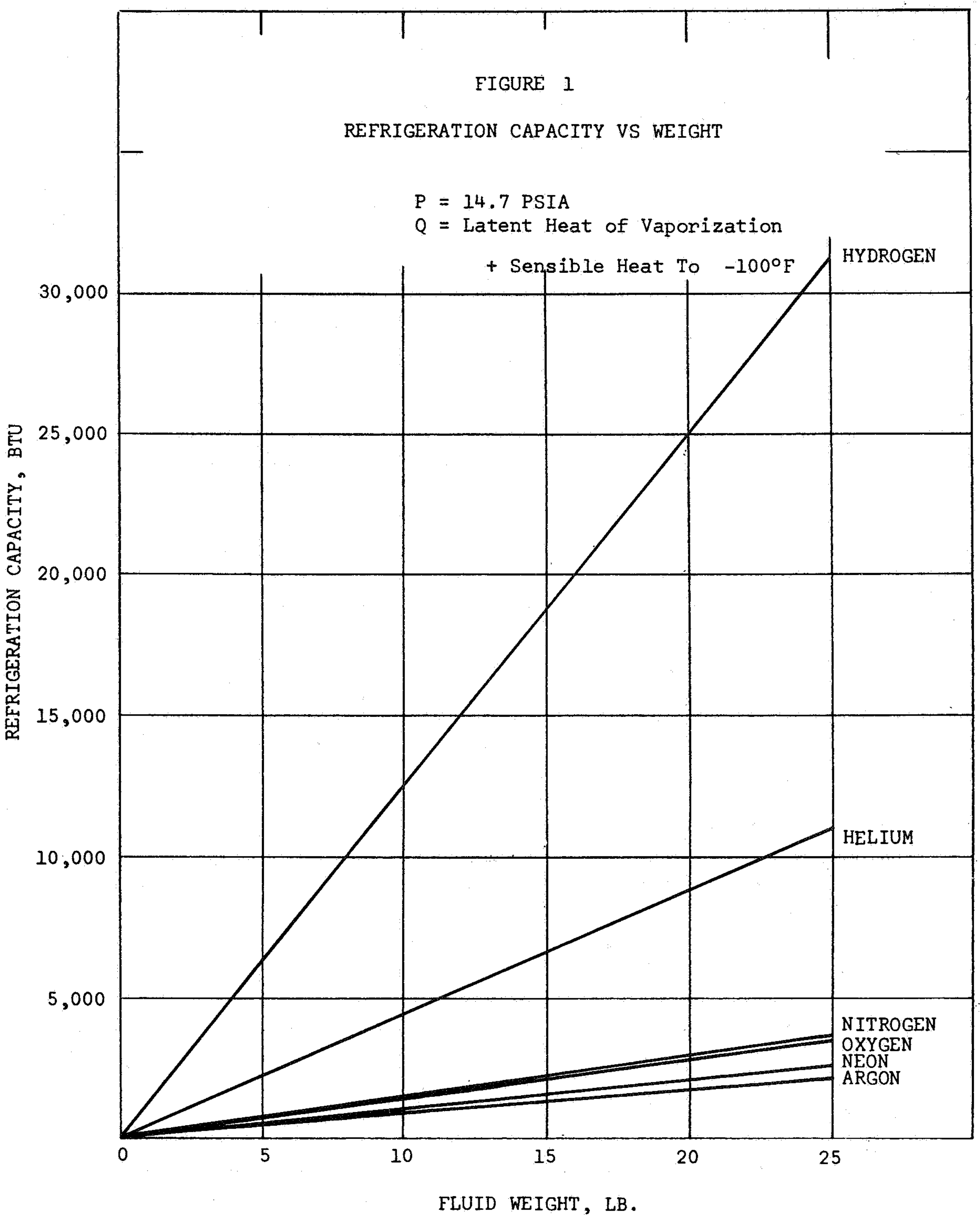
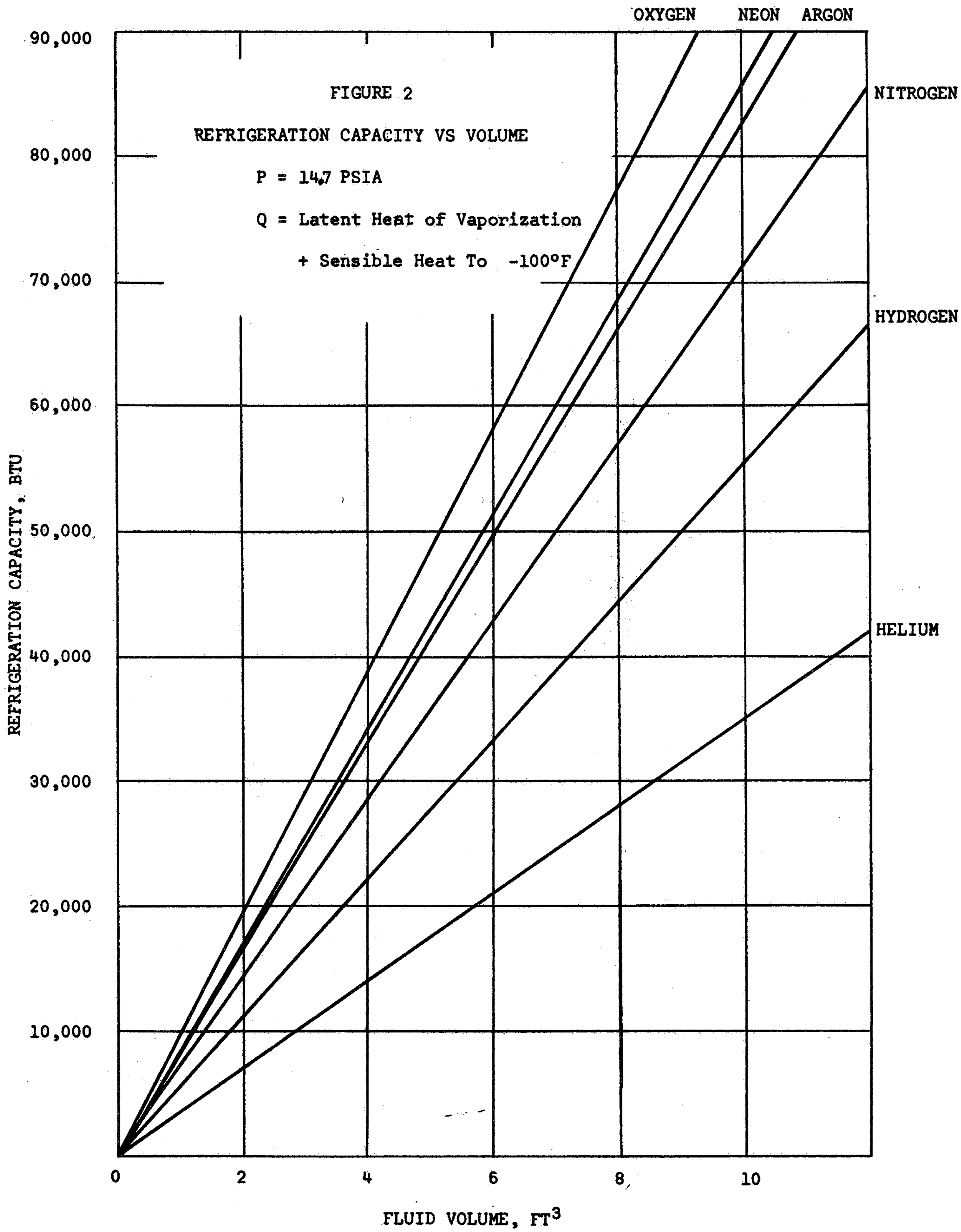


FIGURE 1
REFRIGERATION CAPACITY VS WEIGHT
P = 14.7 PSIA
Q = Latent Heat of Vaporization
+ Sensible Heat To -100°F
HYDROGEN
HELIUM
NITROGEN
OXYGEN
NEON
ARGON
30,000
25,000
20,000
15,000
10,000
5,000
0
5
10
15
20
25
REFRIGERATION CAPACITY, BTU
FLUID WEIGHT, LB.


OXYGEN
NEON
ARGON
NITROGEN
HYDROGEN
HELIUM
FIGURE 2
REFRIGERATION CAPACITY VS VOLUME
P = 14.7 PSIA
Q = Latent Heat of Vaporization
+ Sensible Heat To -100°F
REFRIGERATION CAPACITY, BTU
90,000
80,000
70,000
60,000
50,000
40,000
30,000
20,000
10,000
0
2
4
6
8
10
FLUID VOLUME, FT3

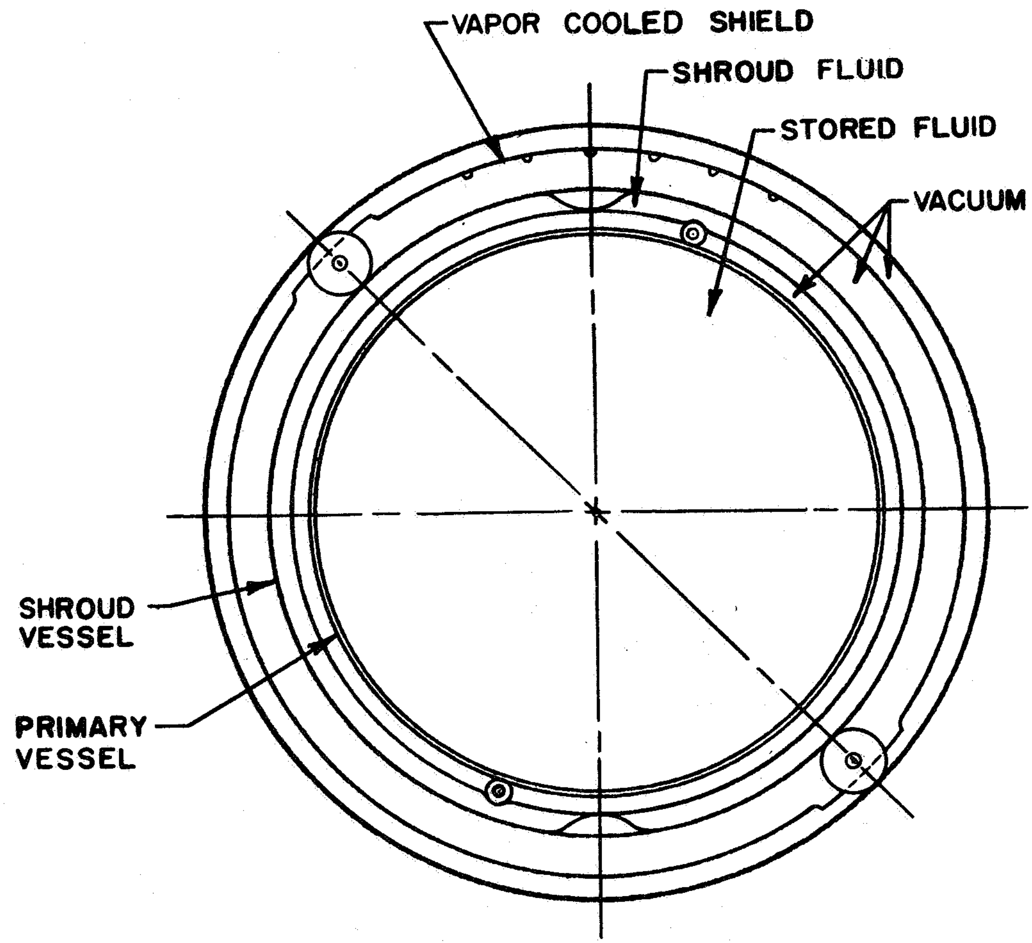


FIGURE 3

ISOTHERMAL-MOUNTED SHROUD DESIGN

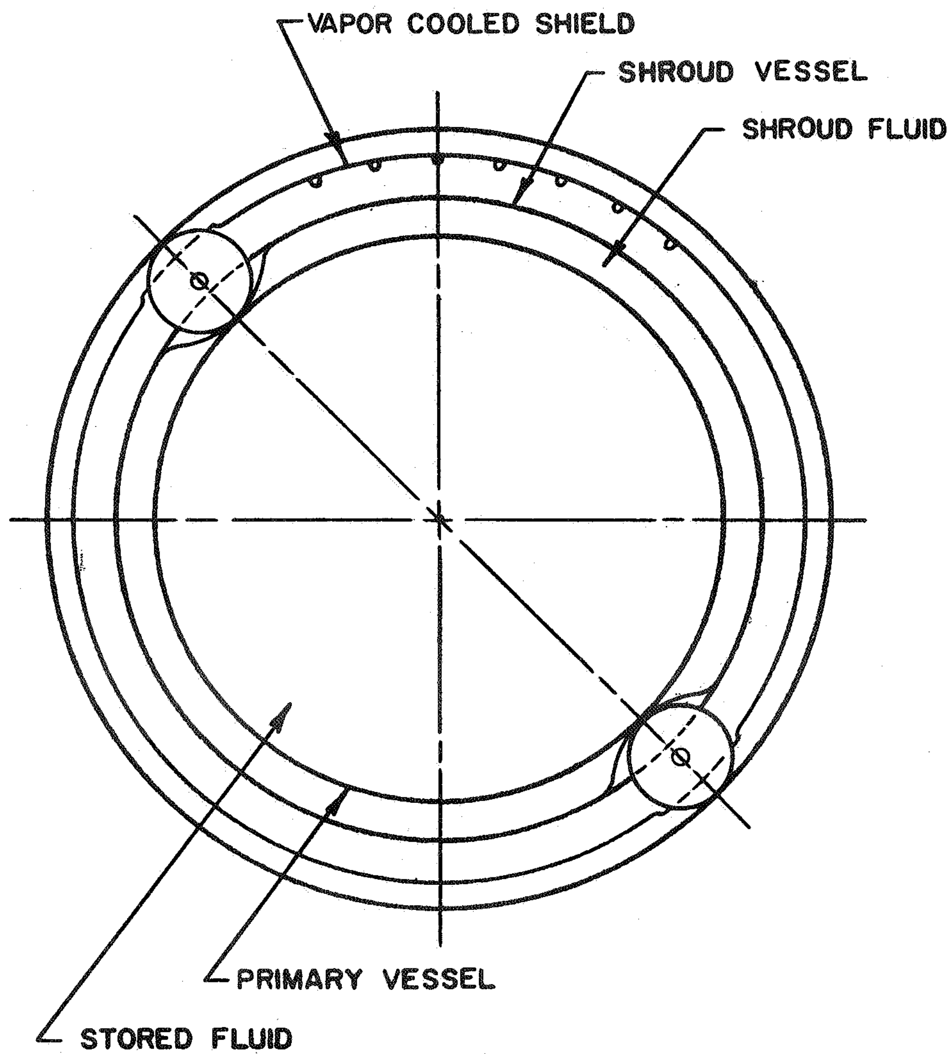


FIGURE 4

INTEGRALLY-MOUNTED SHROUD DESIGN

enclosed in a high vacuum environment. Minimum conductive heat transfer from the shroud vessel to the inner vessel is provided by low thermal-conductivity radial bumpers which support the inner storage vessel within the shroud vessel. Radiant heat transfer to the inner vessel is minimized by the high vacuum annulus between the shroud and inner vessel.

The shroud vessel itself is a completely separate spherical annulus composed of an inner and outer shell for the purpose of storing the secondary fluid. The shroud-inner vessel combination is supported within the outer shell by radial bumpers, and again with a high vacuum annulus between the outer shell and the shroud reducing radiant heat transfer. The vacuum voids between the shroud and the inner vessel, and between the shroud and the outer shell would not necessarily have to be separate, but would be a continuous volume to facilitate fabrication.

Further reduction of radiant heat transfer to both the inner vessel and shroud vessel is accomplished by using discrete shields which are located between the shroud and inner vessel as well as the shroud and outer shell. The flow of vaporized fluid from the shroud vessel is transferred through a vapor-cooling coil attached to one of the radiation shields, and thus provides a lower temperature environment for the shroud fluid. This same method can be used on one of the radiation shields surrounding the inner vessel to utilize the primary fluid for vapor cooling when it is withdrawn from the inner vessel.

The radiation shields, the outer surface of the inner vessel, and the inner and outer surfaces of the shroud vessel are all silver plated to provide low-emissivity surfaces which will minimize radiant heat flow. Since the outer shell is subjected to higher temperatures, the inner surface is copper-plated to obtain minimum emissivity for lower radiant heat transfer to the shroud vessel.

INTEGRALLY-MOUNTED SHROUD DESIGN

As previously stated, the integrally-mounted shroud system differs from the isothermally-mounted system in that the secondary cryogenic fluid physically contacts the outer surface of the inner storage vessel. There is also no vacuum jacket separating the two vessels. The inner vessel is supported within the shroud vessel at the radial bumper locations. These radial bumper locations are inverted cups in the shroud tank, which have been designed and fabricated by a secondary forming process to come in physical contact with the inner vessel. The radial bumpers are located between the shroud unit and the outer shell, and serve to support the shroud unit itself within the outer shell. A high vacuum is maintained within the annulus between the shroud and outer shell. Discrete radiation shields are used, with one shield vapor-cooled by vaporization of the secondary fluid in the shroud. It is also possible to vapor-cool one shield by utilizing the inner vessel fluid; however, this particular design is not shown in the schematic. All surfaces exposed in the vacuum annulus are silver- and copper-plated, as in the isothermally-mounted shroud design, to minimize radiant heat input by providing low-emissivity reflecting surfaces.

Perhaps the most distinct advantage of the isothermal design over the integral design is the reduction of the direct conductive heat input to the inner vessel. The schematic of the integrally-mounted system depicted in Figure 4 shows that conductive paths to the inner vessel from the outer shell exist through the support bumpers and shroud shell. This heat flow is very low however, since the support bumpers are fabricated from low-thermal conductivity material. Both systems do have direct conductive heat paths from the outer shell to the inner vessel through the fill and vent lines, but the long lengths of these lines serve to keep this heat transfer to a minimum.

For the same inner vessel volume in both systems, the isothermally-mounted shroud system would of necessity involve a larger outer shell and increased weight due to the additional support bumpers, longer lines, and the second shroud shell. If this same size outer shell was used for the integrally-mounted system, with an inner vessel size equivalent to the isothermal system, larger support bumpers could be utilized which would increase the path length of the conductive heat flow and thereby reduce the heat transfer. From this standpoint, the integral system would be competitive with the isothermal system, and it would probably remain superior in terms of weight economy.

When comparing the isothermal system with the integral system, it was concluded that the isothermal system presents fabrication and assembly complications, and that these complications were not offset by either weight, size, or thermal advantages. Therefore, because its relatively simple design would result in a more rapid fabrication of a prototype shroud system for this contract, the integrally-mounted shroud system was selected. The final prototype design is similar to that shown in Figure 4, and is discussed more fully later in the report. All analyses of the shroud concept discussed hereafter are based on an integrally-mounted shroud design.

Since the primary stored fluid in the inner storage vessel is separated from the shroud fluid by only the inner vessel wall, it will be assumed in all cases that the two fluids are at the same temperature. Table I shows the melting and boiling temperatures, at 14.7 psia, for the six cryogens examined in Figures 1 and 2.

TABLE I

MELTING AND BOILING POINTS FOR VARIOUS FLUIDS (14.7 psia)

FLUID	MELTING POINT °R	BOILING POINT °R
Oxygen	98.7	162.3
Argon	150.7	157.1
Nitrogen	113.9	139.3
Neon	44.0	48.8
Hydrogen	25.3	36.7
Helium	1.8 @ 368 psia	7.6

When both the shroud fluid and the primary fluid are stored at 14.7 psia, then from Table I various combinations of fluids can be selected such that the primary cryogen is either solid, liquid, or gaseous, depending upon the temperature of the shroud fluid. Other combinations which may be implied from the data are those in which the shroud fluid and/or the primary fluid are pressurized. Further, the pressurized fluids may be stored at pressures above or below their individual supercritical pressures. Due to the complexity of an analysis of all such combinations, only a few will be examined in this report. These analyses will be representative of the effective use of shroud cooling. Storage of solid cryogens has not been considered; however, this particular storage condition is obviously attainable, and may have very definite applications for long term cryogenic storage.

SHROUD AND PRIMARY FLUIDS AT 14.7 PSIA

Certain combinations of the cryogens in Table I are impossible for this application if no solidification is desired. Only nitrogen-oxygen and argon-oxygen combinations are possible.

There are two alternatives for utilizing a liquid nitrogen-liquid oxygen combination, depending upon which is selected as the primary storage fluid. It will be assumed that the primary storage fluid is oxygen, due to its application in environmental control and fuel cell systems. Since the boiling temperature of vented nitrogen is below that of oxygen, a fill procedure would be required that would permit continuous circulation of liquid nitrogen into the shroud until the liquid oxygen has been subcooled to liquid nitrogen temperature. When the system has been completely filled, radiant heat input to the shroud-inner vessel assembly through the high vacuum void will vaporize only nitrogen in the shroud, maintaining a controlled temperature environment for the oxygen. Due to its subcooled state, the liquid oxygen will not vaporize.

Conductive heat input to the system, mainly through the radial support bumpers, could cause some heat transfer directly into the inner vessel. Accompanying this would be a temperature rise within the oxygen. Additional heat will transfer to the oxygen when the liquid level in the shroud declines (in a 1-g environment) and the shroud gas phase surrounds more of the inner vessel. Heat will then be transferred from the shroud to the inner vessel by conduction through the shroud gas phase. As a result of the temperature rise in the oxygen, and with the nitrogen vaporizing at constant temperature, a temperature differential will exist and heat will flow from the oxygen to the nitrogen. Eventually an equilibrium condition will exist, with little or no heat remaining in the oxygen to cause vaporization. If the shroud is maintained charged with liquid nitrogen, very little oxygen loss should result.

Liquid argon can be utilized in the shroud instead of liquid nitrogen as a secondary refrigerant to cool liquid oxygen. In this case, however, the liquid oxygen would be subcooled only 5 R° to equal the temperature of the vaporizing argon, compared with a subcooling of 23 R° to equal liquid nitrogen temperature. It appears that liquid nitrogen would therefore be more suitable for cooling liquid oxygen.

PRIMARY FLUID PRESSURIZED

The most effective use for shroud cooling a primary stored fluid results when the primary fluid is pressurized above atmospheric pressure. Figures 5, 6 and 7 show the effects of pressure on the density of oxygen, hydrogen, and helium, respectively, when stored at various temperatures corresponding to the shroud liquid. Using these data, the storage of oxygen, hydrogen and helium in shrouded systems was examined. A common inner storage vessel of Ti 6Al-4V, with an inner diameter of 26.500 in., was selected for the analysis. Primary fluid storage pressures of 3000, 1500, and 500 psia were chosen. Inner vessel wall thickness was calculated according to the following formula:

$$t = \frac{SF \times P \times D}{4 \times UTS}$$

A UTS for Ti 6Al-4V of 140,000 psi at 530°R and a safety factor of 2 were utilized in the calculations. No factor was used in the calculations to account for the creep characteristics of Ti 6Al-4V at cryogenic temperatures. If used, this factor would decrease the material efficiency to about 75%, thus resulting in a larger wall thickness and vessel weight. The data presented in this section are therefore useful for comparison purposes only, and do not represent actual system weights. Inner vessel weight was based upon a density of 0.16 lb/in^3.

Standby time was based upon the time required for all shroud liquid to vaporize and expand to -100°F at 14.7 psia. The amount of heat required per unit volume was taken directly from the data presented in Figure 2. The rate of heat input to the shroud fluid was assumed to be dependent upon the shroud liquid temperature, based upon accumulated data from numerous cryogenic storage dewars manufactured and tested at Bendix. The heat input rates presented in Table II were used in the analysis.

TABLE II

HEAT INPUT RATES FOR VARIOUS SHROUD FLUIDS
PURE VACUUM INSULATION

SHROUD FLUID	Q HEAT INPUT BTU/hr-ft^2 SHROUD
Oxygen	1.2
Argon	1.2
Nitrogen	1.2
Neon	1.0
Hydrogen	1.0

Standby time was therefore determined from the following:

$$\theta = \frac{Q' V_s}{Q A_s}$$

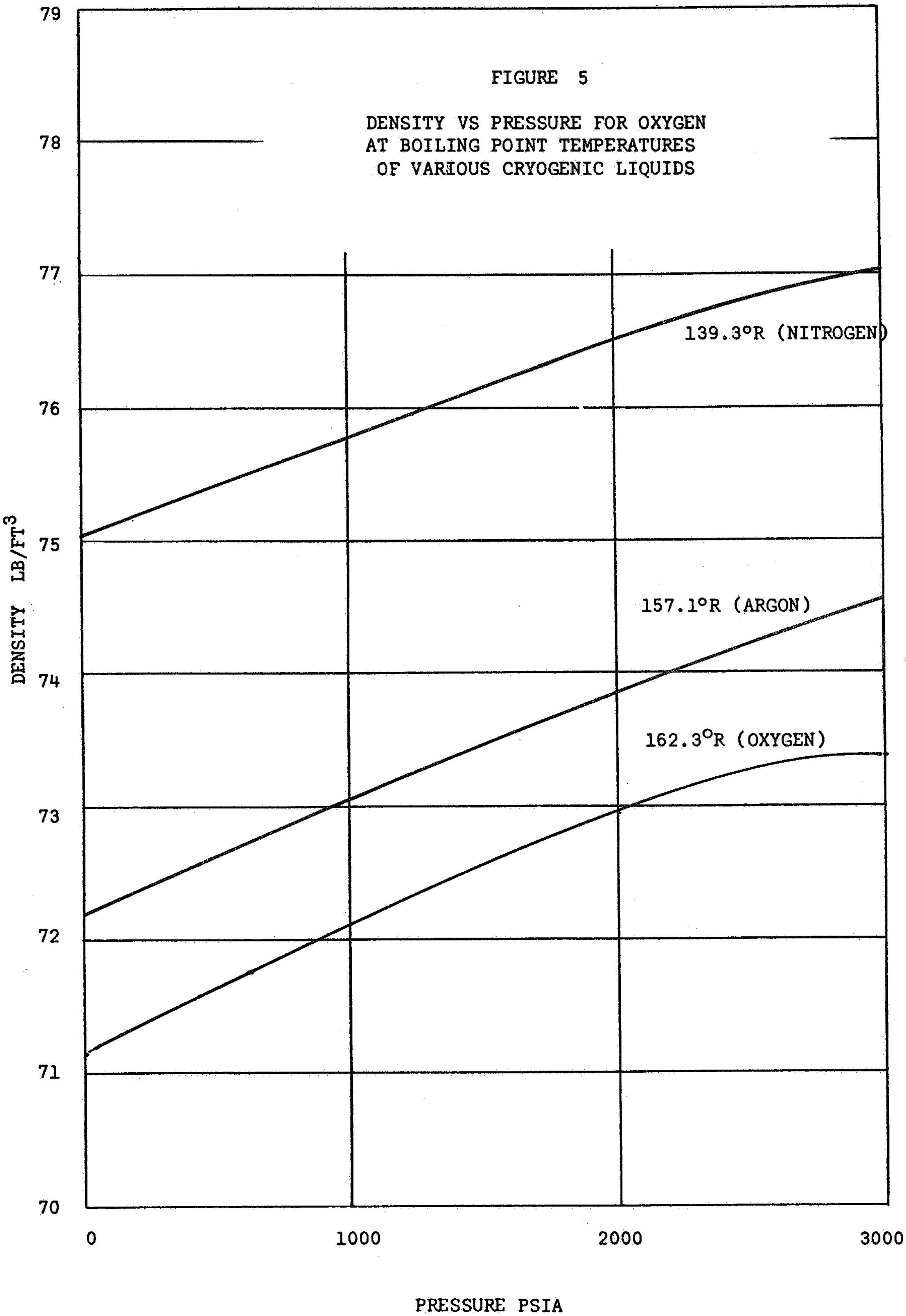


FIGURE 5
DENSITY VS PRESSURE FOR OXYGEN
AT BOILING POINT TEMPERATURES
OF VARIOUS CRYOGENIC LIQUIDS
139.3°R (NITROGEN)
157.1°R (ARGON)
162.3°R (OXYGEN)
DENSITY LB/FT3
79
78
77
76
75
74
73
72
71
70
0
1000
2000
3000
PRESSURE PSIA

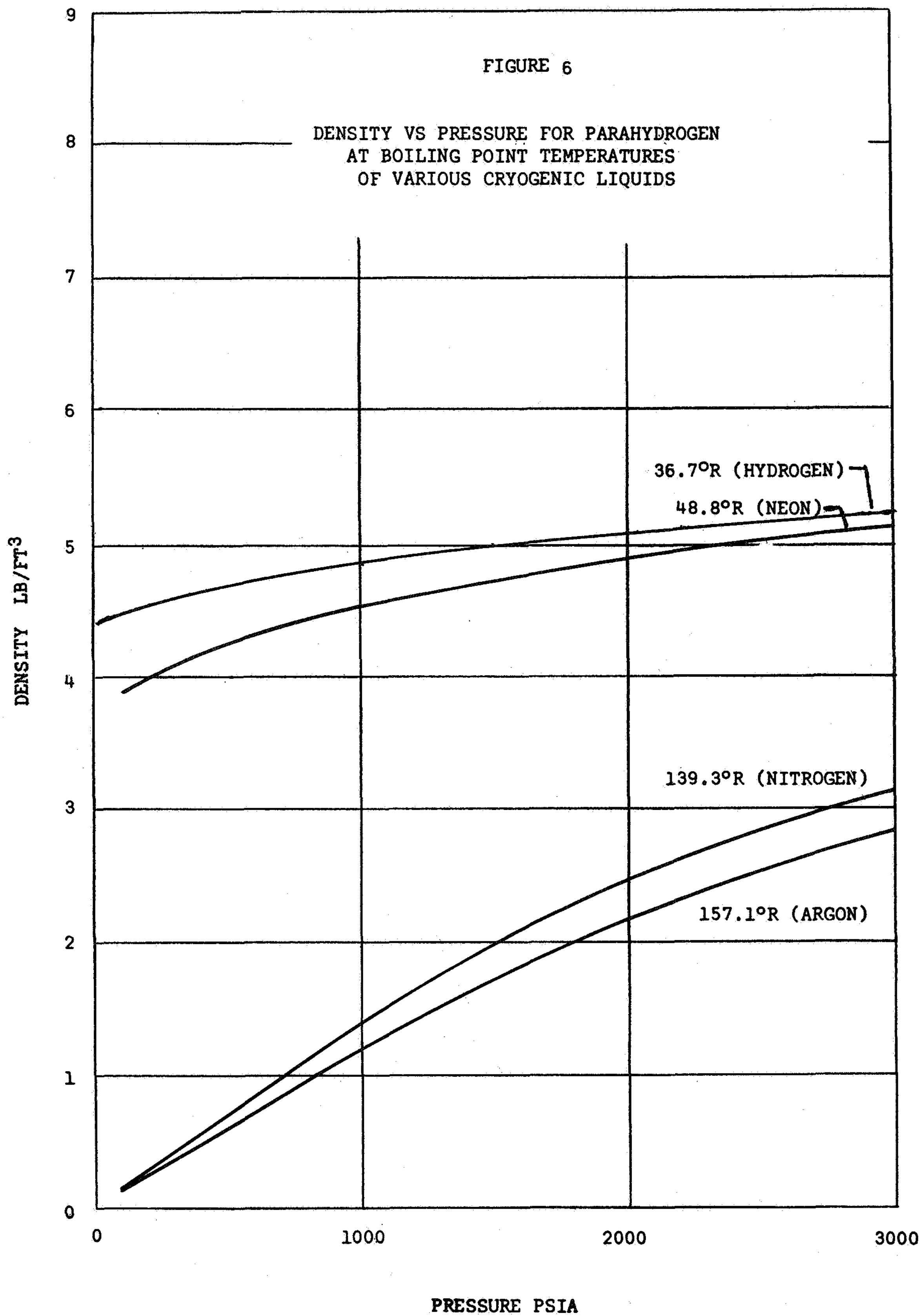


FIGURE 6
DENSITY VS PRESSURE FOR PARAHYDROGEN
AT BOILING POINT TEMPERATURES
OF VARIOUS CRYOGENIC LIQUIDS
36.7°R (HYDROGEN)
48.8°R (NEON)
139.3°R (NITROGEN)
157.1°R (ARGON)
DENSITY LB/FT3
PRESSURE PSIA
0
1
2
3
4
5
6
7
8
9
0
1000
2000
3000

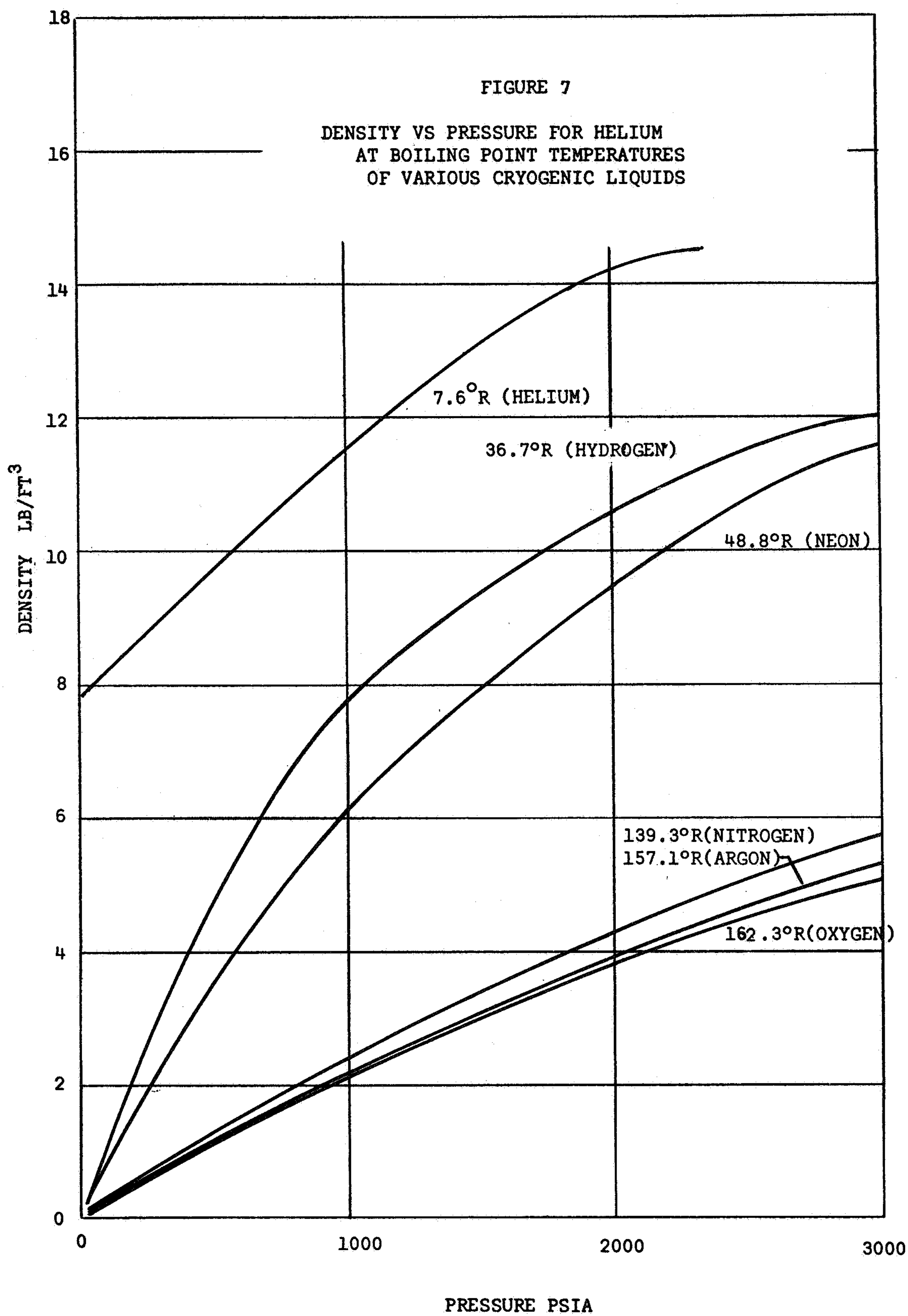


FIGURE 7
DENSITY VS PRESSURE FOR HELIUM
AT BOILING POINT TEMPERATURES
OF VARIOUS CRYOGENIC LIQUIDS
7.6°R (HELIUM)
36.7°R (HYDROGEN)
48.8°R (NEON)
139.3°R(NITROGEN)
157.1°R(ARGON)
162.3°R(OXYGEN)
DENSITY LB/FT3
PRESSURE PSIA
0
2
4
6
8
10
12
14
16
18
0
1000
2000
3000

Variation in standby time was accomplished by changing the diameter of the shroud vessel, thus varying the volume of shroud liquid. The Ti 6Al-4V shroud wall thickness was calculated from the following modified Zoelly equation:

$$t = D\sqrt{\frac{SF \times P \times \sqrt{3(1-\gamma^2)}}{4 \times Y \times 0.214}}$$

A safety factor of 2, modulus of elasticity of 17.5×10^6 psi, Poisson's ratio of 0.3, and external pressure of 14.7 psia were used in the calculations.

Outer shell wall thickness was determined from the modified Zoelly buckling equation. 304L stainless steel was selected as outer shell material, resulting in a modulus of elasticity of 29×10^6 psi and Poisson's ratio of 0.3. A safety factor of 1.5 and external pressure of 14.7 psia were used in the calculations. Outer shell weight was based upon a density of 0.29 lb/in^3.

Both pre-launch and inflight cooling of oxygen, hydrogen and helium were examined. Figures 8 through 10 show the effects of inner storage pressure and shroud liquid on the dry-shroud system weight and standby time. The plotted ratio "Dry-Shroud System Weight/Primary Fluid Weight" consists of the ratio of the sum of the outer shell, shroud vessel, inner vessel, and primary fluid weights to the primary fluid weight. This ratio is applicable, therefore, to a system in which the shroud is "dry", or completely empty, at launch, but with no primary fluid loss during the pre-launch (ground) standby period. Additional system weight components such as fittings, lines, support bumpers, etc., were not included. This addition, however, will be nearly constant for all systems; therefore the figures show trends and not actual system weight ratios.

The results presented in Figure 8 show that of the three pressures examined, an oxygen system at the subcritical pressure of 500 psia and cooled by liquid nitrogen is lightest in weight for pre-launch cooling. Argon cooling results in practically the same results, with the difference being the lower density of oxygen at liquid argon temperature. Neon cooling of hydrogen at 500 psia was found to be the most weight economical, as shown in Figure 9. It is interesting to note that for neon cooling, system weight decreases with decreasing hydrogen pressure, while for nitrogen and argon cooling, system weight decreases with increasing hydrogen pressure. Oxygen was not considered as a shroud refrigerant for hydrogen.

Pre-launch cooling of helium stored at 1500 psia and cooled by liquid hydrogen is shown in Figure 10 to result in the lightest system weight. Both 500 and 3000 psia helium storage pressures result in higher system weights, for both hydrogen and neon shroud cooling. Since the analysis was limited to three primary storage pressures, no optimization was intended in this examination. The results indicate that a minimum system weight can be obtained with hydrogen cooling of helium at some pressure between 1500 and 3000 psia.

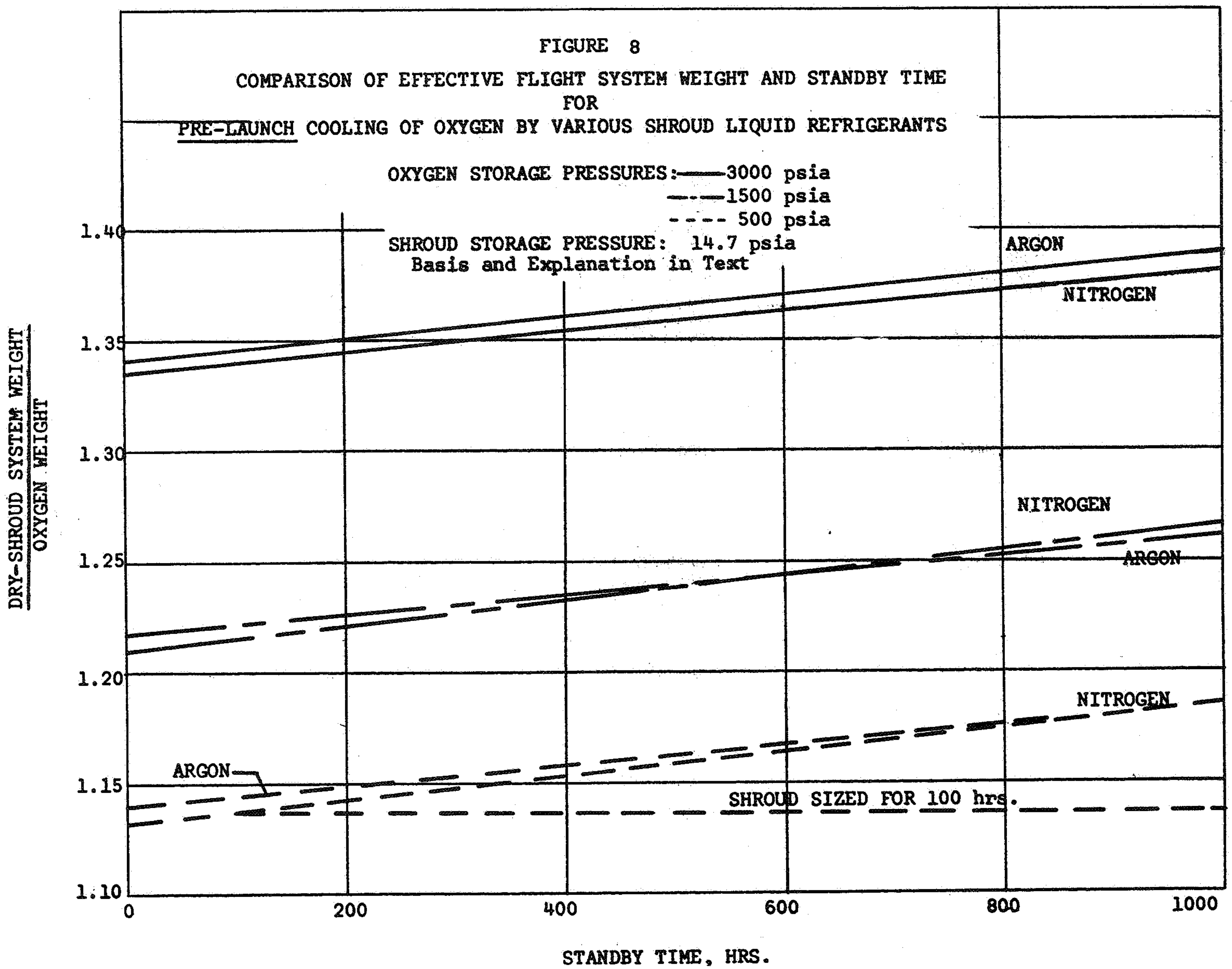


FIGURE 8

COMPARISON OF EFFECTIVE FLIGHT SYSTEM WEIGHT AND STANDBY TIME FOR PRE-LAUNCH COOLING OF OXYGEN BY VARIOUS SHROUD LIQUID REFRIGERANTS

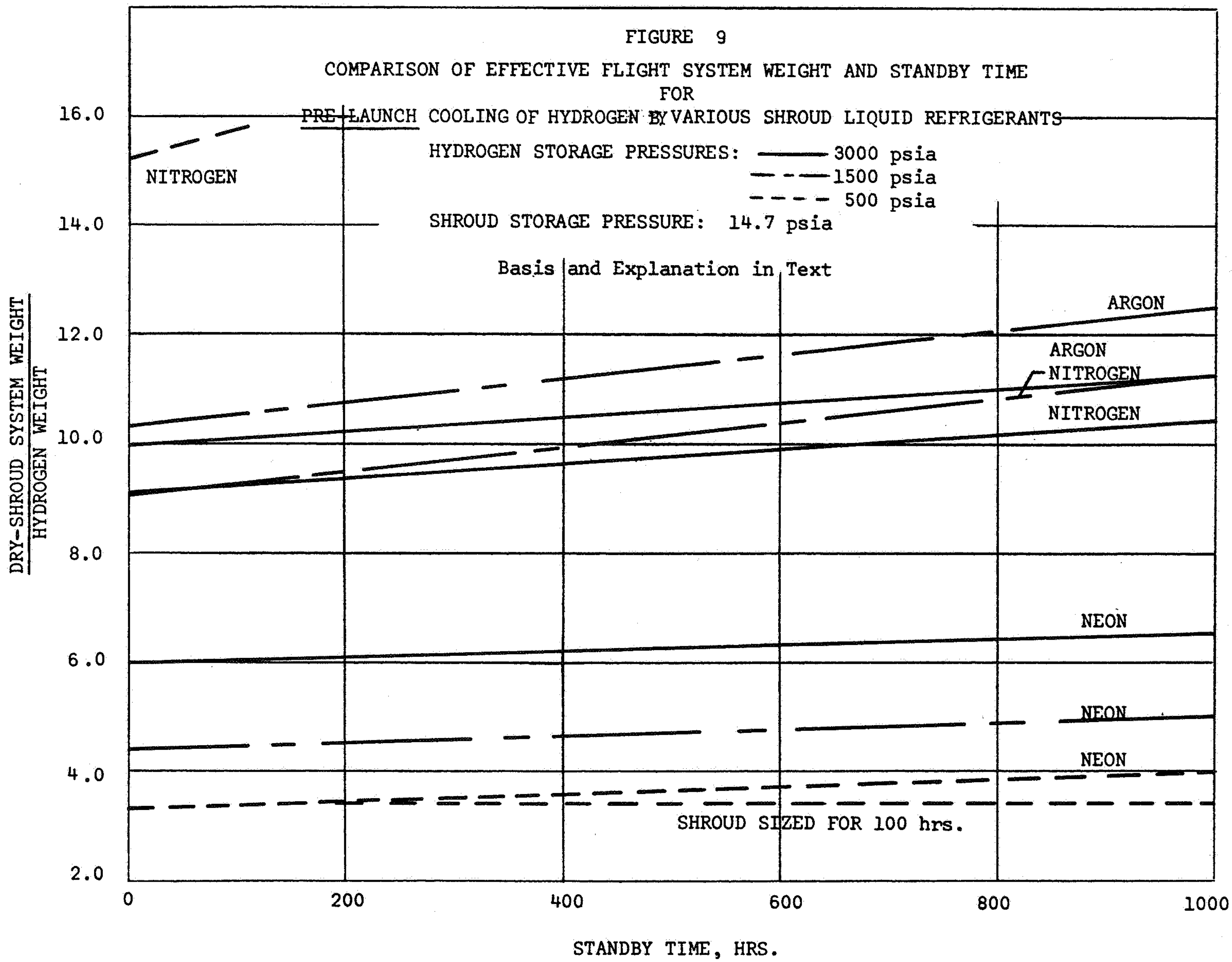


FIGURE 9

COMPARISON OF EFFECTIVE FLIGHT SYSTEM WEIGHT AND STANDBY TIME
FOR
PRE-LAUNCH COOLING OF HYDROGEN BY VARIOUS SHROUD LIQUID REFRIGERANTS

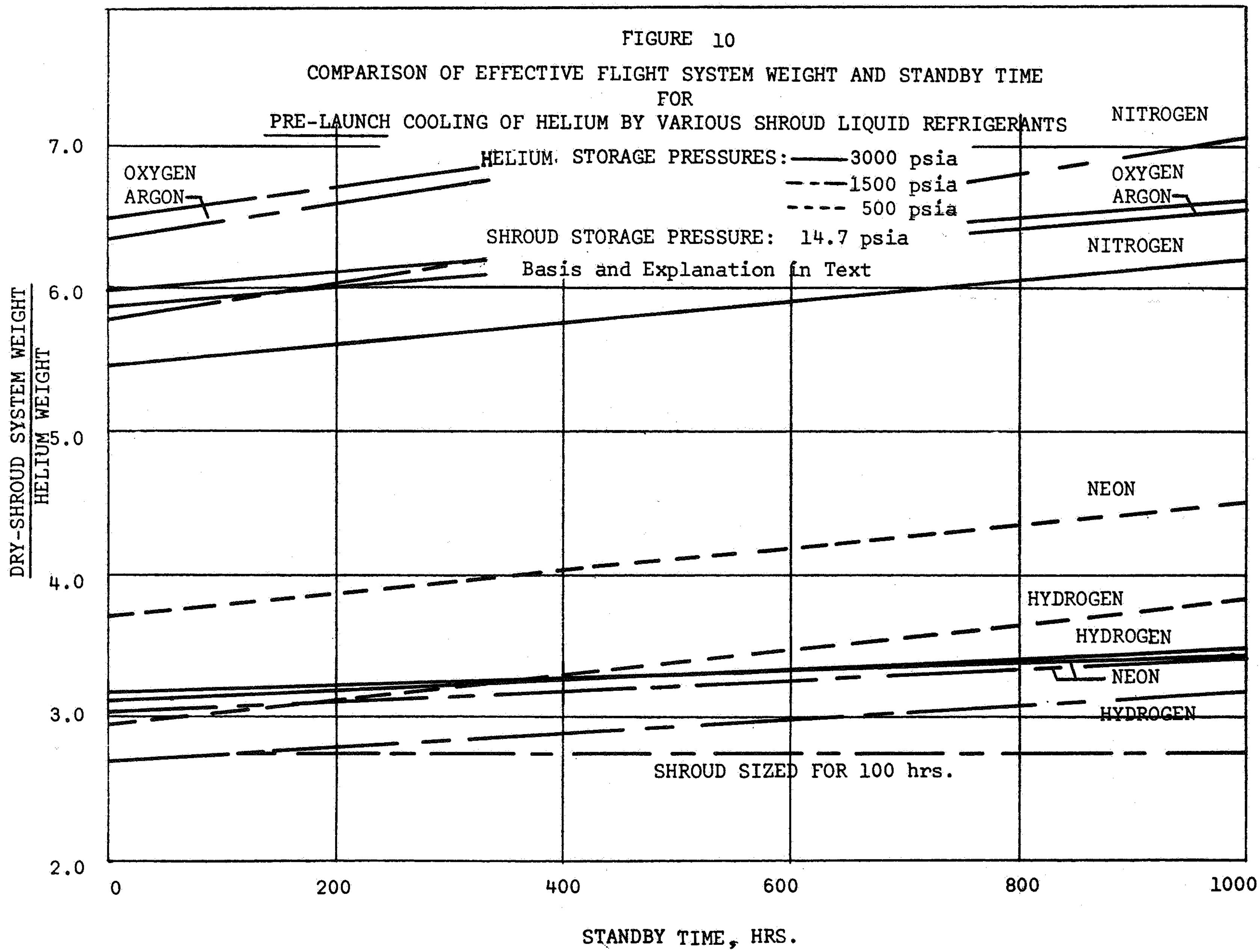


FIGURE 10

COMPARISON OF EFFECTIVE FLIGHT SYSTEM WEIGHT AND STANDBY TIME FOR PRE-LAUNCH COOLING OF HELIUM BY VARIOUS SHROUD LIQUID REFRIGERANTS

It was anticipated that neon would be more effective than hydrogen as a shroud refrigerant for helium. This assumption was based upon its larger cooling capacity (see Figure 2), relatively low temperature, and the fact that its density does not affect the dry-shroud system weight. However, the lower temperature of liquid hydrogen permits a higher helium storage density. Thus, this increased storage weight of the primary fluid offsets the increased volume and weight required for a hydrogen shrouded system to produce standby times comparable to those attained by liquid neon. Figure 10 depicts this condition.

As previously stated, standby times were varied by increasing the shroud volume. Refilling of the shroud system is easily accomplished, and therefore it would not be weight economical to design the shroud for one filling when pre-launch cooling is its primary function. Therefore, a system could be designed based upon 50-100 hours standby time, and if standby is continued beyond that time, the shroud could be refilled as often as is necessary. The effect of shroud sizing for 100 hours standby is shown for the optimum fluids in Figures 8 through 10.

Inflight cooling of the three primary fluids is examined in Figures 11 through 13. The "Wet-Shroud System Weight" consists of the sum of the outer shell, shroud vessel, shroud fluid, inner vessel, and primary fluid weights. This ratio is applicable to a system in which the shroud is completely filled (and venting) at launch. These figures therefore show the effects of the shroud fluid densities on flight system weights.

The minimum system weights for storage of the three fluids resulted from the same combination of primary storage pressure and shroud fluid as noted in the pre-launch cooling curves. Based upon the maximum standby period studied, 1000 hours, nitrogen cooling of oxygen at 500 psia was most weight economical for inflight oxygen cooling (See Figure 11). Figure 12 shows that neon cooling of 500 psia hydrogen results in the lightest system weight. It was again noted that hydrogen system weight decreases with decreasing hydrogen storage pressure for neon cooling, whereas system weight decreases with increasing storage pressure for nitrogen and argon cooling. Helium stored at 1500 psia and cooled by liquid hydrogen resulted in the lightest flight system, as shown in Figure 13. The effect of the high neon density is shown on these curves. Systems cooled by liquid hydrogen do not show weight changes as rapidly as the neon-cooled systems.

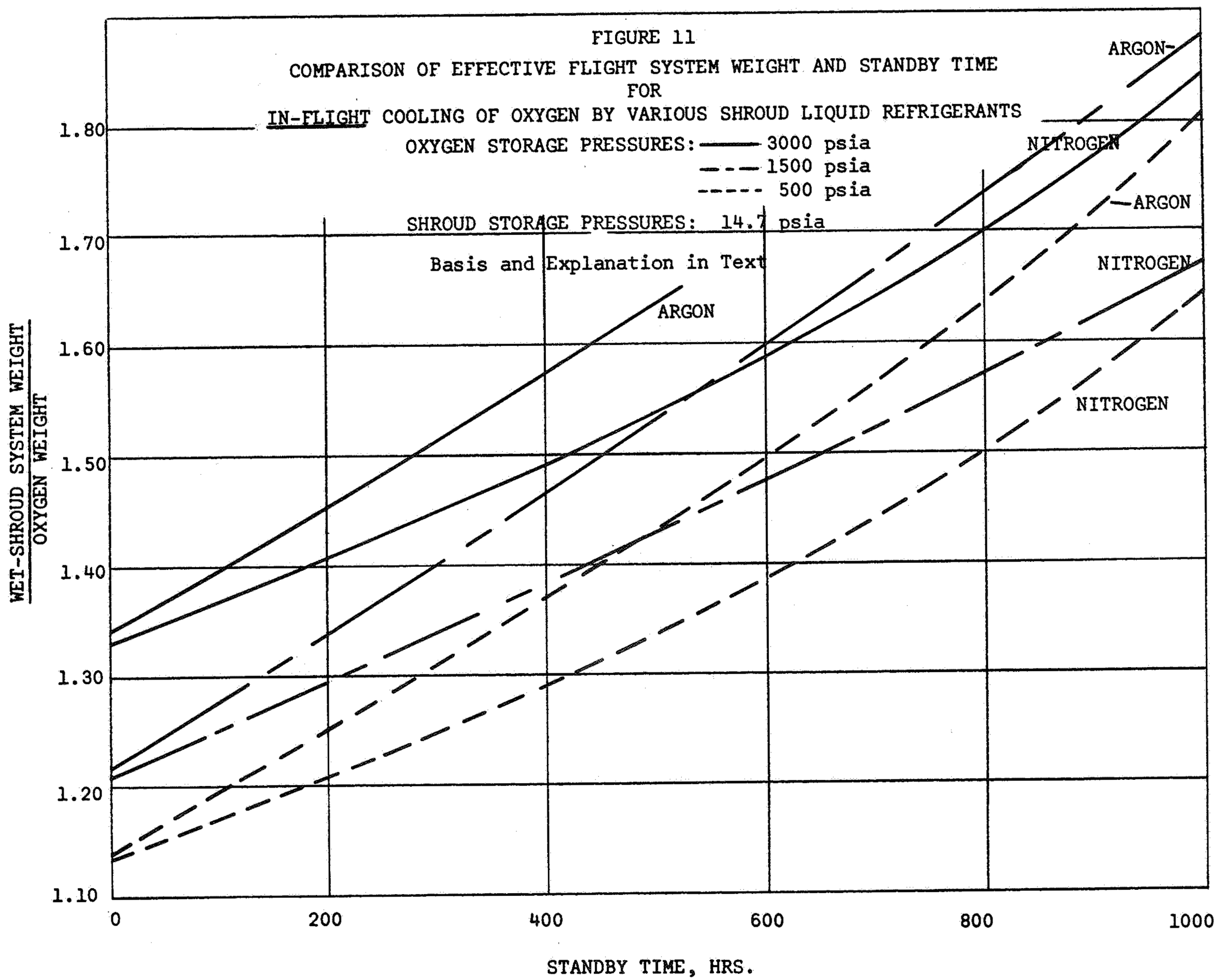


FIGURE 11
COMPARISON OF EFFECTIVE FLIGHT SYSTEM WEIGHT AND STANDBY TIME
FOR
IN-FLIGHT COOLING OF OXYGEN BY VARIOUS SHROUD LIQUID REFRIGERANTS
OXYGEN STORAGE PRESSURES: 3000 psia
1500 psia
500 psia
SHROUD STORAGE PRESSURES: 14.7 psia
Basis and Explanation in Text
ARGON
NITROGEN
ARGON
NITROGEN
ARGON
NITROGEN
WET-SHROUD SYSTEM WEIGHT / OXYGEN WEIGHT
1.10
1.20
1.30
1.40
1.50
1.60
1.70
1.80
0
200
400
600
800
1000
STANDBY TIME, HRS.

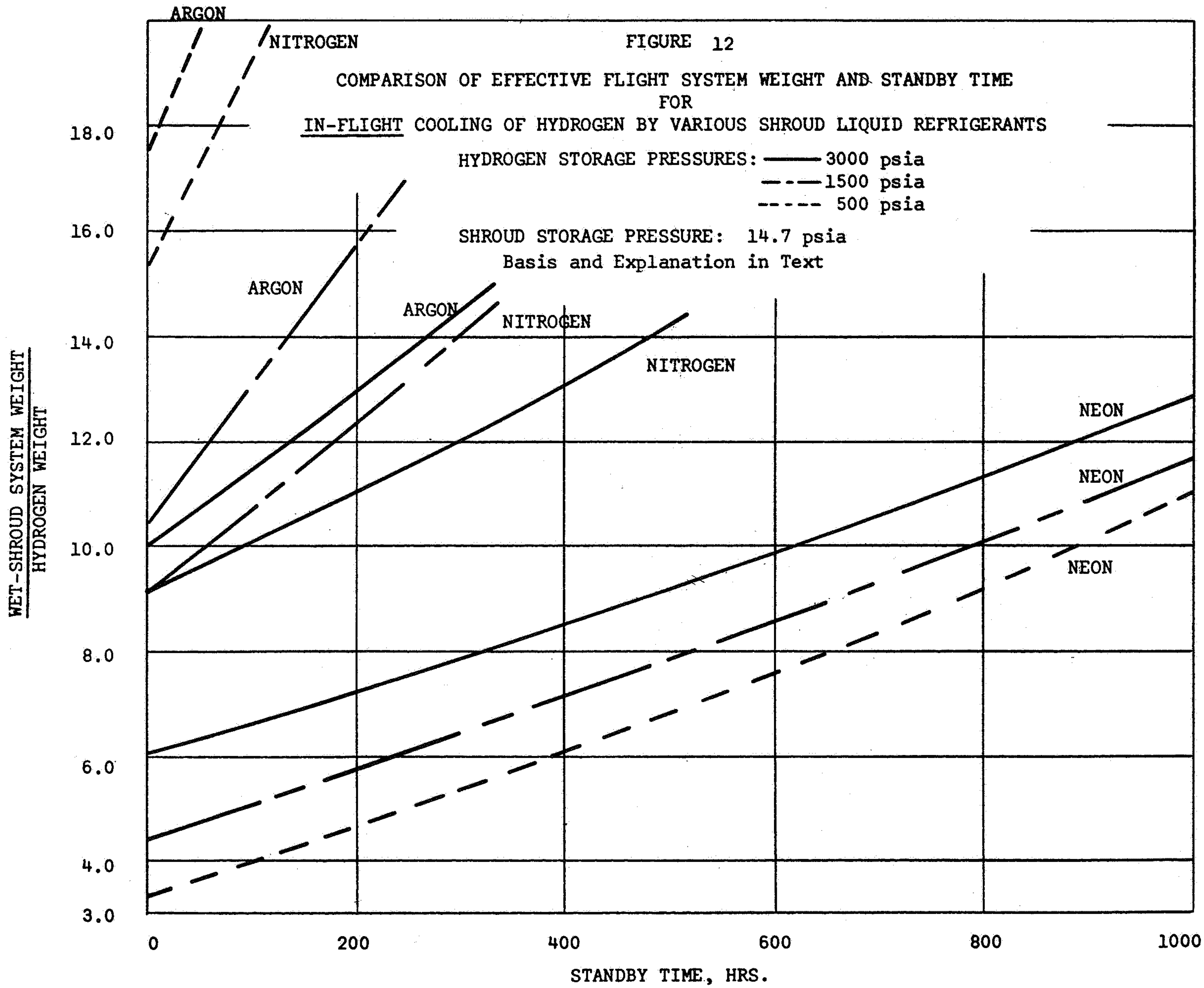


FIGURE 12

COMPARISON OF EFFECTIVE FLIGHT SYSTEM WEIGHT AND STANDBY TIME
FOR
<u>IN-FLIGHT</u> COOLING OF HYDROGEN BY VARIOUS SHROUD LIQUID REFRIGERANTS

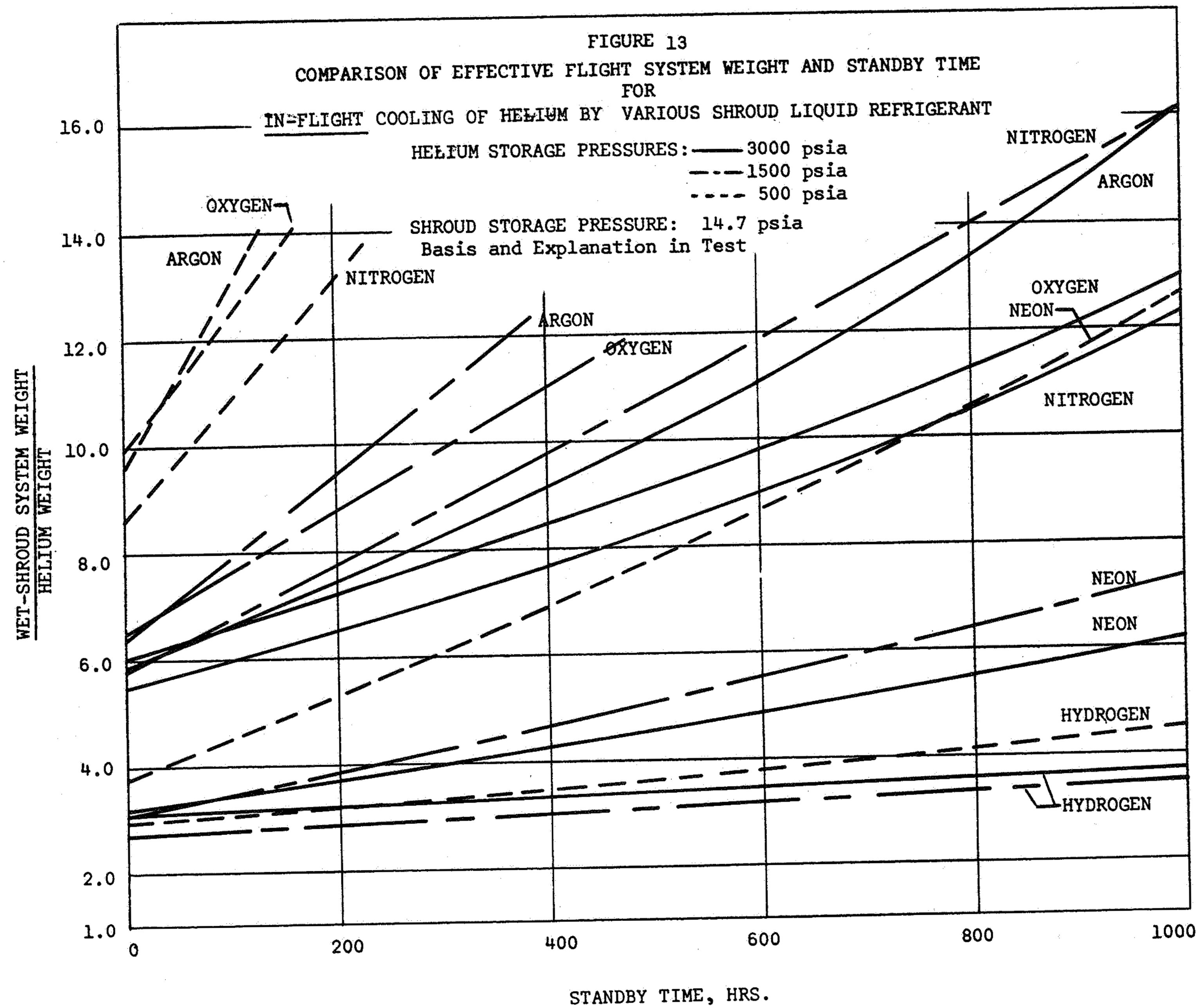


FIGURE 13
COMPARISON OF EFFECTIVE FLIGHT SYSTEM WEIGHT AND STANDBY TIME
FOR
IN-FLIGHT COOLING OF HELIUM BY VARIOUS SHROUD LIQUID REFRIGERANT

INSULATION EFFECTS

The results presented in Figures 8 through 13 were based upon a shroud storage vessel with pure vacuum insulation. Heat input to the stored cryogens may be further reduced by utilization of additional insulation techniques, thus increasing standby time. Insulation techniques examined include laminar insulation, discrete radiation shields, and vapor-cooled shields.

LAMINAR INSULATION

Laminar insulation is composed of numerous alternate layers of radiation shielding material and low conductivity spacing material. The usual method of application to cryogenic storage vessels consists of building up the insulating layers on the outside surface of the inner storage vessel, which is in turn enclosed within a high vacuum environment. For comparison purposes, Linde Sl-62 insulation was selected as a typical example of laminar insulation to be used for insulating a shroud-inner vessel assembly. The following thermal conductivity values, obtained from the best data available, were used as a basis for this exercise. Their values are based on flat plate data.

TABLE III

THERMAL CONDUCTIVITY FOR LINDE Sl-62 INSULATION
NON-COMPRESSED, FLAT PLATE
BETWEEN 530°R AND TEST TEMPERATURE

TEST TEMPERATURE °R	THERMAL CONDUCTIVITY BTU/hr-ft^2 °R/ft
36.7	2.7×10^{-5}
162.3	2.2×10^{-5}

Tests conducted with superinsulating materials wrapped around spherical containers have indicated that thermal conductivity is greater than that obtained with flat plate tests. These effects are more apparent with smaller diameter vessels than with larger spheres. To account for these discrepancies, the conductivity values presented in Table III were multiplied by a factor of 3, which more closely results in the heat input rates expected for a storage unit of the size studied. It was assumed that the superinsulation was attached to the outer surface of the shroud vessel, and completely filled the evacuated void between the shroud and outer vacuum jacket. Only the weight-optimum combinations of storage pressure and shroud fluid given in Figures 8 through 13 were used to compare the various insulating techniques examined in this section. Three shroud fluids were therefore used, with liquid nitrogen cooling oxygen at 500 psia storage pressure; liquid neon cooling hydrogen at 500 psia, and liquid hydrogen cooling helium at 1500 psia. Table IV presents the modified thermal conductivity values used for superinsulating the shroud-inner vessel combinations.

TABLE IV

THERMAL CONDUCTIVITY FOR LINDE S1-62 INSULATION
NON-COMPRESSED, FTC = 3
BETWEEN 530°R and SHROUD FLUID TEMPERATURE

SHROUD FLUID	Q THERMAL CONDUCTIVITY BTU/hr-ft^2 °R/ft
Nitrogen	8.1×10^{-5}
Neon	6.6×10^{-5}
Hydrogen	6.6×10^{-5}

Standby time for the superinsulated shroud units was determined from the following:

$$\theta = \frac{Q' \, Vs}{Q \, A_{si}}$$

Two superinsulation thicknesses, 0.5 inch and 1.0 inch, were used. Insulation weight was based on a density of 4.7 lb/ft^3.

DISCRETE RADIATION SHIELD INSULATION

A discrete radiation shield is a low emissivity radiation barrier placed in the vacuum space and isothermally-mounted to some support structure. At equilibrium, it assumes some intermediate temperature between the inner and outer vessel surface temperatures. The heat transfer between the surfaces in discrete shielded vessels is effected by solid conduction through supports and interconnecting lines, thermal transfer by the residual gas, and thermal radiation.

In a high vacuum insulated vessel, with well designed supports and plumbing, more than half of the heat transfer to the cryogenic fluid is the result of radiation. When a dewar has established equilibrium heat transfer, the temperatures of the surfaces remain constant and the emissivity factor limits the rate of radiant heat transfer. The use of discrete shields with good thermal isolation in the vacuum space will substantially reduce the radiant heat transfer. For example, when the emissivity factors are the same between all pairs of surfaces in a dewar, and shields are thermally isolated, the thermal transfer rate to the inner tank may be expressed as:

$$Q = \sigma A \frac{E}{(n+1)} (T_2^4 - T_1^4)$$

n = number of discrete shields

The heat-input rates to the shroud fluid for one- and two-shielded systems are presented in Table V. These values are based on accumulated data from various cryogenic storage dewars manufactured and tested at Bendix.

TABLE V

HEAT INPUT RATES FOR VARIOUS SHROUD FLUIDS
DISCRETE RADIATION SHIELD INSULATION

SHROUD FLUID	NO. SHIELDS	Q HEAT INPUT BTU/hr-ft^2 SHROUD
Nitrogen	1	0.8
Nitrogen	2	0.6
Neon	1	0.75
Neon	2	0.5
Hydrogen	1	0.75
Hydrogen	2	0.5

Standby time for the shielded units was determined from the following:

$$\theta = \frac{Q' V_s}{Q A_s}$$

Insulation weights were based on Mg shields, with 0.015 in. wall thickness. Spacing and location of shields within the vacuum void were predicated on fabrication and assembly limitations.

VAPOR-COOLED SHIELDS

A vapor-cooled shield is a discrete radiation shield that is cooled by the effluent fluid from the tank it is shielding. This reduces the temperature below that of a simple discrete shield and provides a more effective radiation shield. The same requirements that apply to simple discrete shields also apply to vapor-cooled shields in that they are thermally isolated from the tank and adjacent shields, and there is an absence of gas between the shields.

In practice, fluid issuing from a dewar inner tank is routed either through a double walled discrete shield, or through tubes or channels attached to a discrete shield. Efficient heat exchange between the fluid and shield will be accomplished before the fluid reaches the normal vent or supply line to the outside of the dewar. The fluid enters the shield at the same temperature as the fluid in either the inner tank or shroud, depending upon which fluid is being utilized, and leaves the shield at the shield temperature if the heat exchange is complete. Therefore, heat reaching a vapor-cooled shield is divided into two portions. One portion is that which is absorbed by the fluid passing through and carried outside the dewar, and the other portion passes on to the inner tank (or shroud vessel, if applicable). The vapor-cooled shield will come to a lower equilibrium temperature, and therefore the heat transferred to the inner tank (or shroud) will be less than that of a non vapor-cooled shield. The heat absorbed by the fluid as it passes through the vapor-cooled shield will be equal to its change in enthalpy within the shield if pressure remains constant. Considering the heat transfer between surfaces to be composed of components of radiation and conduction, the following relations may be applied to dewars with a vapor-cooled shield.

The rate of heat transfer (Q_1) to a vapor-cooled shield from enclosing surfaces is:

$$Q_1 = R_1 + C_1,$$

where R_1 and C_1 are radiation and conduction components.

Since the heat arriving at the cooled shield is partly absorbed by the effluent fluid, the rate of heat transfer (Q_2) from a vapor-cooled shield to enclosed surfaces may be expressed as:

$$Q_2 = R_2 + C_2,$$

$$= Q_1 - Q_3,$$

where R_2 and C_2 are radiation and conduction components and Q_3 is the rate at which heat is absorbed by the effluent fluid. Q_3 depends upon the rate of mass flow of fluid from the inner container and its change in specific enthalpy as it passes through the vapor cooled shield:

$$Q_3 = \dot{m}\,\Delta h.$$

It follows that:

$$Q_2 = Q_1 - \dot{m}\,\Delta h.$$

The application of vapor-cooled shielding to the shroud tank system consists of utilizing the vented shroud fluid for the vapor-cooling function.

Heat input rates to the shroud fluid for a simple vapor-cooled shield and for two shields with one being vapor-cooled are presented in Table VI. Again, these values are based on data accumulated from test programs at Bendix.

TABLE VI

HEAT INPUT RATES FOR VARIOUS SHROUD FLUIDS
VAPOR-COOLED RADIATION SHIELD INSULATION

SHROUD FLUID	NO. SHIELDS	NO. V-C SHIELDS	Q HEAT INPUT BTU/hr-ft^2
Nitrogen	1	1	0.60
Nitrogen	2	1	0.30
Neon	1	1	0.50
Neon	2	1	0.25
Hydrogen	1	1	0.50
Hydrogen	2	1	0.25

Standby time was determined by the same calculations as those used for discrete radiation shields. Insulation weights were increased for the vapor-cooling function by assuming the use of 1/4" stainless-steel tubing spaced at 3.2 feet of tubing per square foot of shield surface area.

Figures 14 through 16 represent the results of applying various insulation techniques to the optimum pre-launch shroud cooling systems shown in Figures 8 through 10. All three figures show that pure vacuum-insulated systems have relatively good standby times, without the additional weight of super insulation or discrete shields.

When liquid nitrogen is used to cool oxygen at 500 psia (Figure 14), system weight advantages for pure vacuum insulation are obtained for pre-launch standby times up to 385 hours. Figure 15 shows that with neon cooling of hydrogen at 500 psia that vacuum insulation is the weight-optimum technique for pre-launch standby time up to 415 hours. For pre-launch standby time of up to 260 hours, pure vacuum insulation provides the lightest system weight when using liquid hydrogen to cool helium at 1500 psia.

Variations in standby times for the shrouded units of Figures 14 through 16 were obtained by increasing the shroud volume, as in Figures 8 through 10. It has previously been shown that the shroud volume can be sized for only 50-100 hours standby time and refilled with shroud fluid if pre-launch standby time is increased. The effect of a shroud sized for 50 hours standby is shown on the vacuum-insulated shroud systems in Figures 14 through 16. From this data it is apparent that for dry-shroud system weight optimization, no insulation techniques are needed in addition to pure vacuum insulation. That is, for maximum ground standby capability with minimum system weight at launch, the optimum shroud system should be sized for some reasonable standby time and should be solely vacuum insulated.

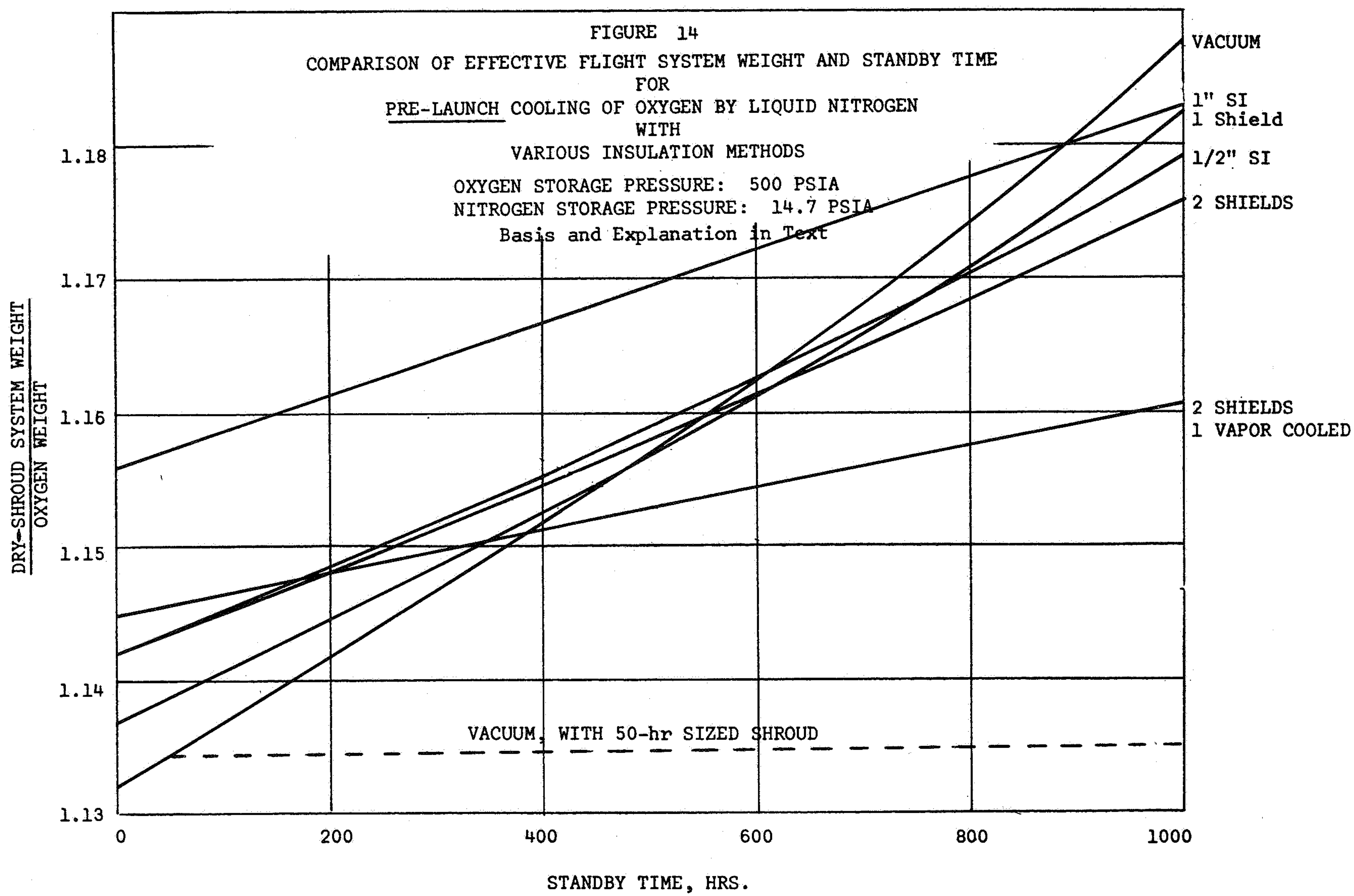


FIGURE 14

COMPARISON OF EFFECTIVE FLIGHT SYSTEM WEIGHT AND STANDBY TIME
FOR
PRE-LAUNCH COOLING OF OXYGEN BY LIQUID NITROGEN
WITH
VARIOUS INSULATION METHODS

OXYGEN STORAGE PRESSURE: 500 PSIA
NITROGEN STORAGE PRESSURE: 14.7 PSIA
Basis and Explanation in Text

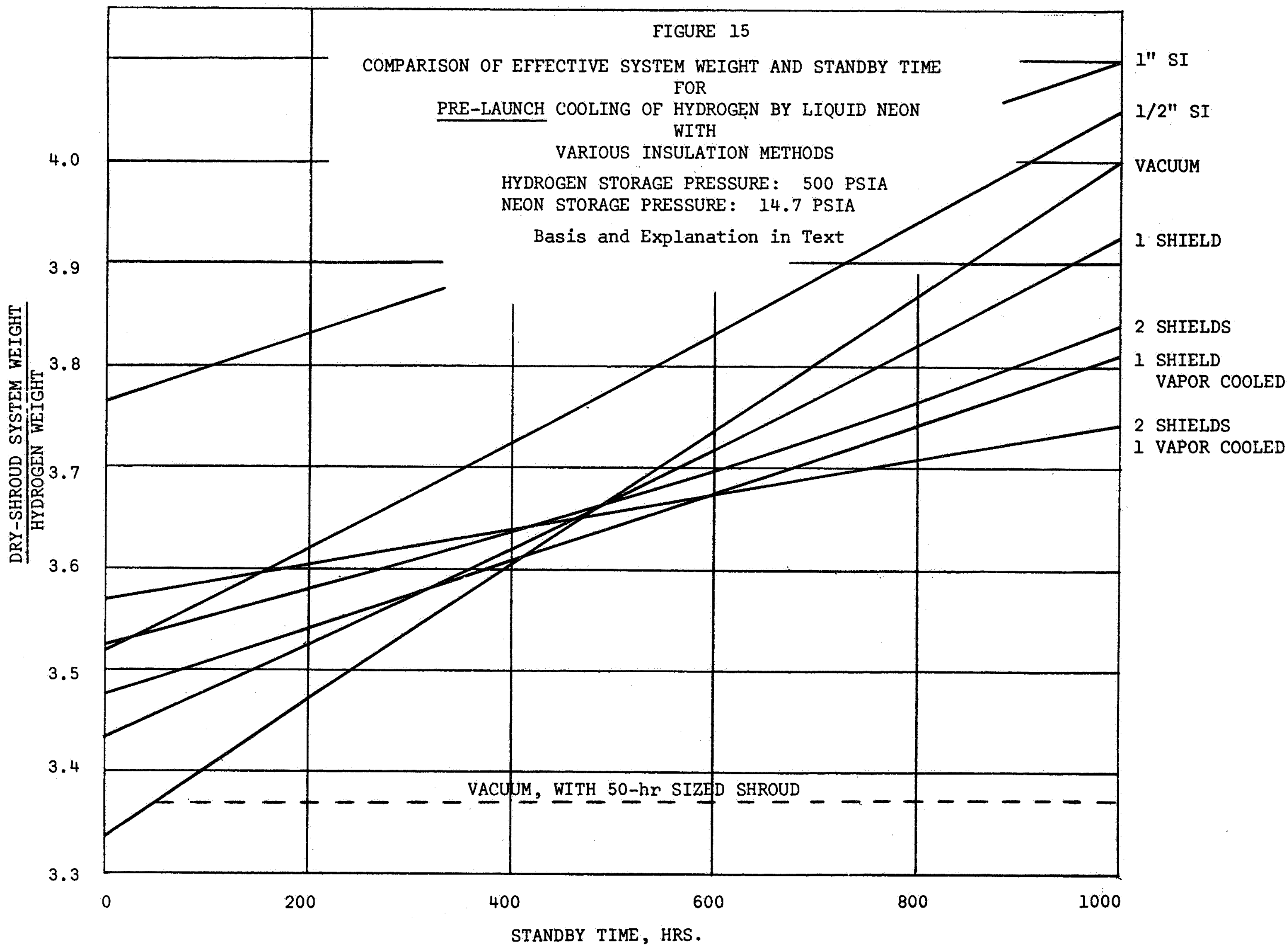


FIGURE 15

COMPARISON OF EFFECTIVE SYSTEM WEIGHT AND STANDBY TIME
FOR
PRE-LAUNCH COOLING OF HYDROGEN BY LIQUID NEON
WITH
VARIOUS INSULATION METHODS

HYDROGEN STORAGE PRESSURE: 500 PSIA
NEON STORAGE PRESSURE: 14.7 PSIA

Basis and Explanation in Text

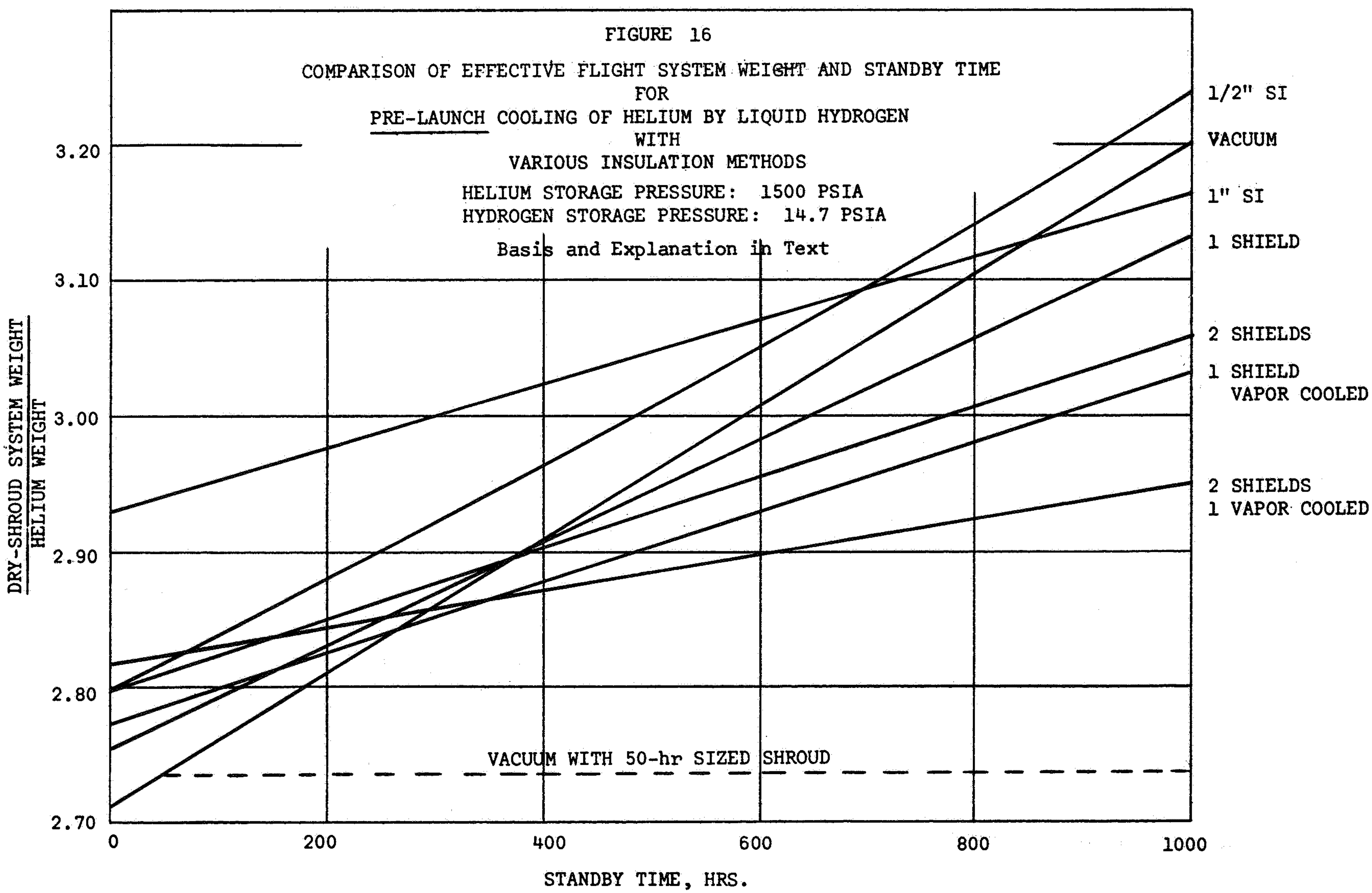


FIGURE 16
COMPARISON OF EFFECTIVE FLIGHT SYSTEM WEIGHT AND STANDBY TIME
FOR
PRE-LAUNCH COOLING OF HELIUM BY LIQUID HYDROGEN
WITH
VARIOUS INSULATION METHODS
HELIUM STORAGE PRESSURE: 1500 PSIA
HYDROGEN STORAGE PRESSURE: 14.7 PSIA
Basis and Explanation in Text
1/2" SI
VACUUM
1" SI
1 SHIELD
2 SHIELDS
1 SHIELD
VAPOR COOLED
2 SHIELDS
1 VAPOR COOLED
VACUUM WITH 50-hr SIZED SHROUD
DRY-SHROUD SYSTEM WEIGHT / HELIUM WEIGHT
3.20
3.10
3.00
2.90
2.80
2.70
0
200
400
600
800
1000
STANDBY TIME, HRS.

For an actual system application, wherein the primary storage fluid must be insulated in-flight as well as during ground standby, the results presented in Figures 14 through 16 may be somewhat misleading. With a dry or empty shroud at launch it has been shown that pure vacuum insulation is most effective for system weight and pre-launch standby optimization. However, pure vacuum insulation would be inadequate for extended standby times during flight. The relative weights and insulating capabilities of laminar insulation and discrete vapor-cooled radiation shields shown in Figures 14 through 16 indicate that the discrete radiation shield concept with vapor-cooling by the primary storage fluid would be optimum. Therefore, in a practical application wherein the shroud fluid is expended at launch, discrete vapor-cooled radiations shields would be utilized for overall system optimization.

Figures 17 through 19 represent the application of various insulating techniques to the optimum in-flight shroud cooling systems discussed in Figures 11 through 13. Pure vacuum insulation advantages are not realized to the degree noted in dry-shroud systems. For liquid nitrogen cooling of oxygen at 500 psia storage pressure, the two-shield insulating mechanism with one shield vapor-cooled by nitrogen is shown to be optimum after 50 hours standby (Figure 17). The same insulating technique is weight and standby optimum for liquid neon cooling of hydrogen at 500 psia for standby times in excess of 60 hours (Figure 18). Figure 19 shows that pure vacuum insulation is weight-advantageous for in-flight cooling of helium at 1500 psia by liquid hydrogen, for standby times up to 185 hours. The two-shield system, with one shield being vapor-cooled by hydrogen, proves to be optimum for longer standby times.

Sizing of the shroud volume for in-flight cooling must be based upon the expected standby time during flight. For ground standby systems, it has been shown that the shroud volume could be limited to some reasonable value, with refilling procedures increasing standby as required. During flight, however, the shroud fluid must be stored in the system at launch. Standby times were therefore varied in Figures 17 through 19 by changing the volume of the shroud.

A comparison of the shroud cooling technique with conventional storage procedures for in-flight applications is presented in Figures 20 through 22. The three concepts considered are (1) shroud cooling systems, (2) conventional non-venting systems, and (3) conventional venting systems.

Standby time for the shroud cooling systems is considered to be the time period between launch (with shroud full) and the time when all shroud fluid has vaporized and expanded to -100°F. All shroud systems employ the two-shield insulating technique, with one shield being vapor-cooled by the shroud fluid. Standby time is varied by changing the volume capacity of the shroud vessel.

For conventional non-venting systems, standby time is the period between filling the storage vessel and the time that the storage vessel pressure attains that value utilized with the comparable shroud system primary vessel pressure. Standby time is increased by increasing the number of discrete radiation shields within the vacuum annulus.

The conventional venting system requires that the storage vessel be initially filled with a quantity of liquid that is greater than the required

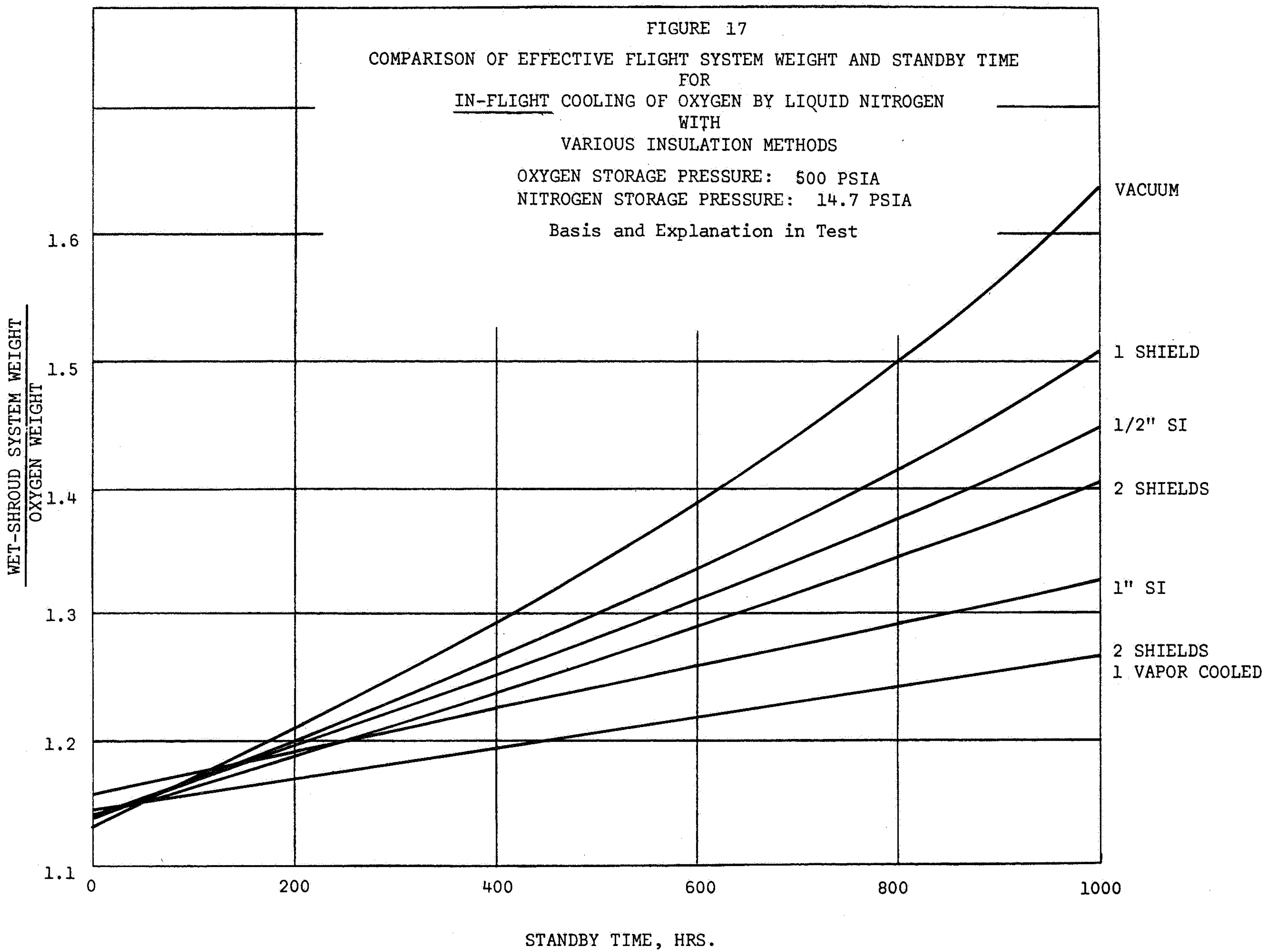


FIGURE 17

COMPARISON OF EFFECTIVE FLIGHT SYSTEM WEIGHT AND STANDBY TIME
FOR
IN-FLIGHT COOLING OF OXYGEN BY LIQUID NITROGEN
WITH
VARIOUS INSULATION METHODS

OXYGEN STORAGE PRESSURE: 500 PSIA
NITROGEN STORAGE PRESSURE: 14.7 PSIA

Basis and Explanation in Test

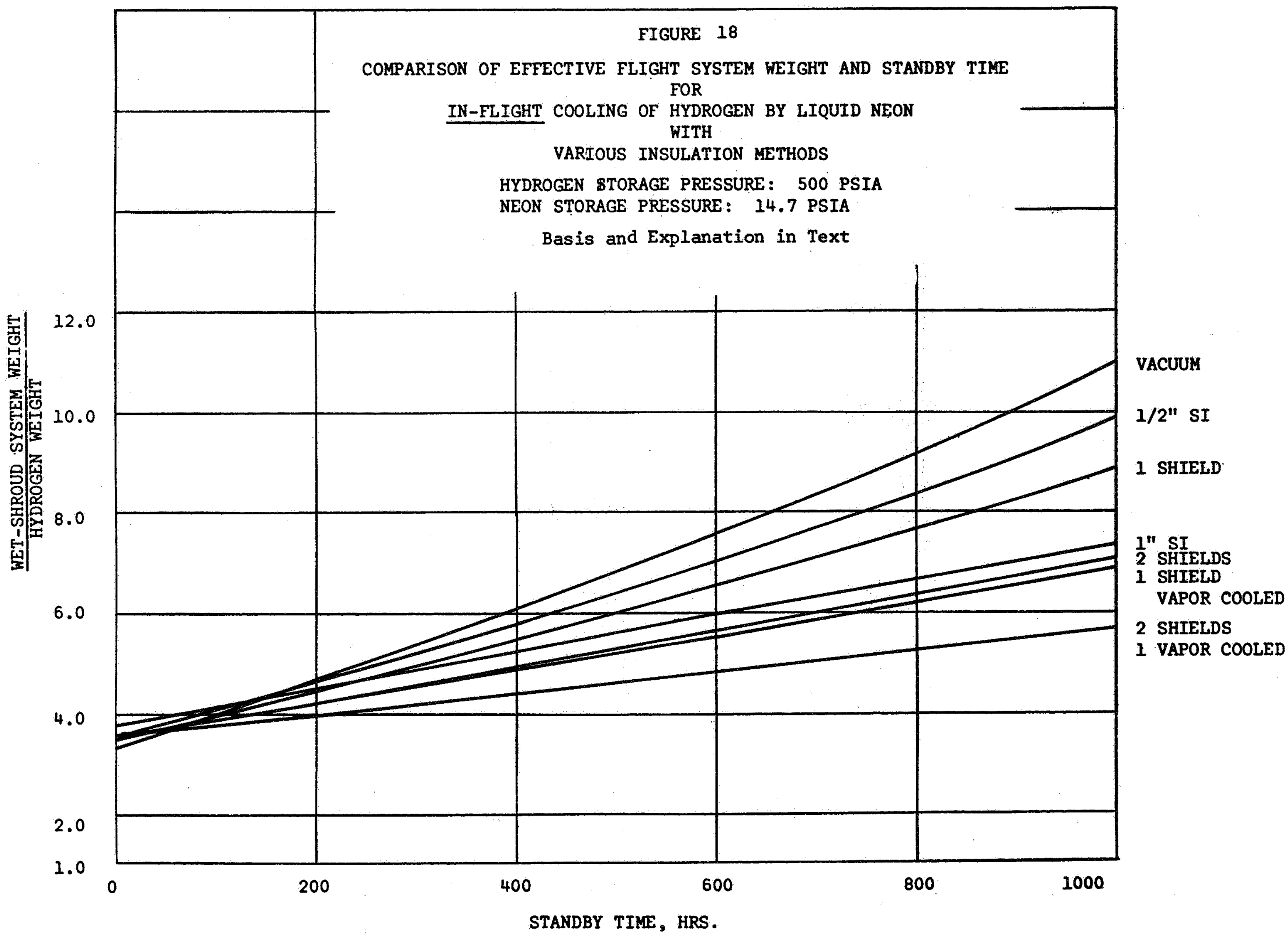


FIGURE 18

COMPARISON OF EFFECTIVE FLIGHT SYSTEM WEIGHT AND STANDBY TIME
FOR
IN-FLIGHT COOLING OF HYDROGEN BY LIQUID NEON
WITH
VARIOUS INSULATION METHODS

HYDROGEN STORAGE PRESSURE: 500 PSIA
NEON STORAGE PRESSURE: 14.7 PSIA

Basis and Explanation in Text

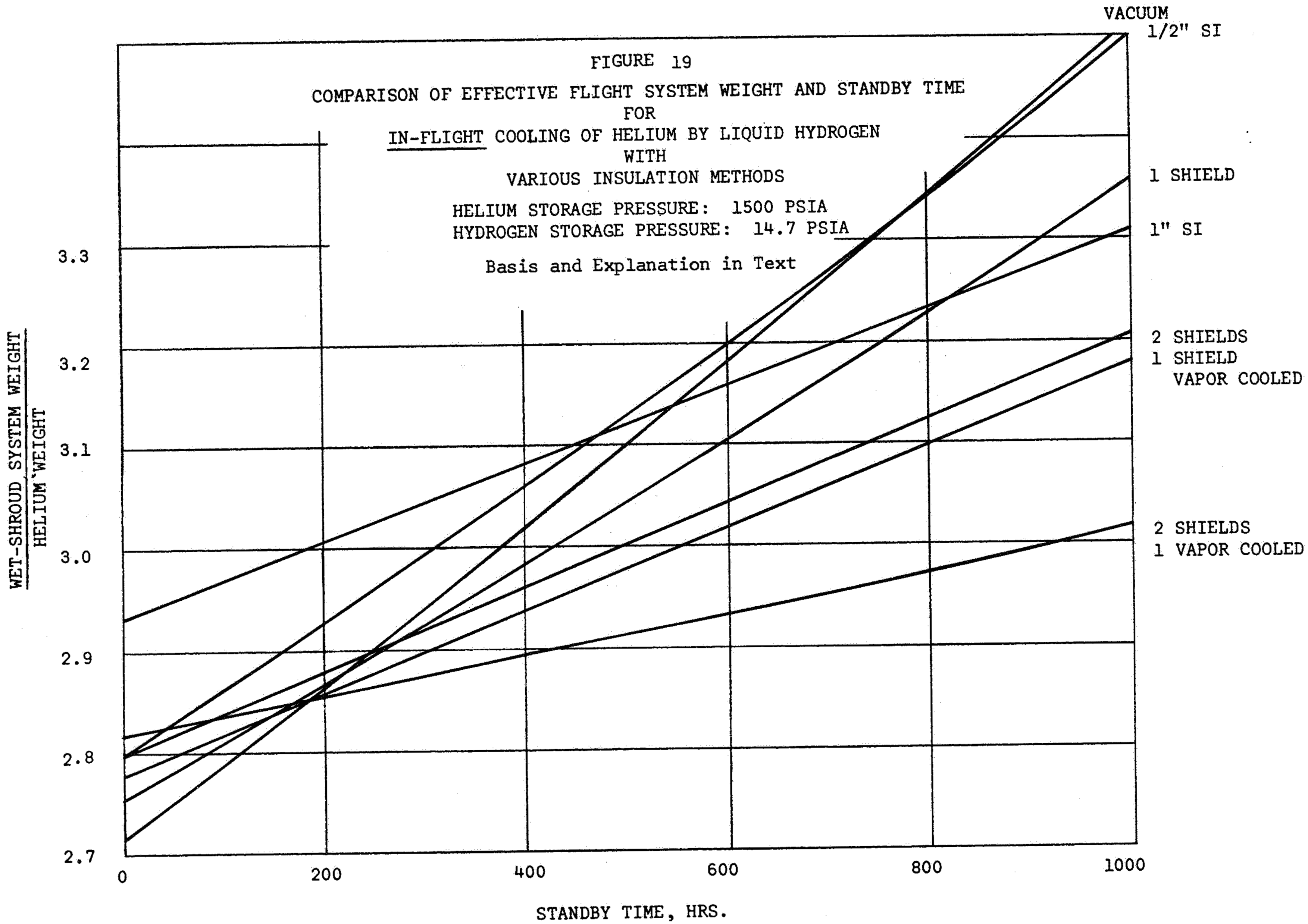


FIGURE 19
COMPARISON OF EFFECTIVE FLIGHT SYSTEM WEIGHT AND STANDBY TIME
FOR
IN-FLIGHT COOLING OF HELIUM BY LIQUID HYDROGEN
WITH
VARIOUS INSULATION METHODS
HELIUM STORAGE PRESSURE: 1500 PSIA
HYDROGEN STORAGE PRESSURE: 14.7 PSIA
Basis and Explanation in Text
VACUUM
1/2" SI
1 SHIELD
1" SI
2 SHIELDS
1 SHIELD
VAPOR COOLED
2 SHIELDS
1 VAPOR COOLED
WET-SHROUD SYSTEM WEIGHT
HELIUM WEIGHT
3.3
3.2
3.1
3.0
2.9
2.8
2.7
0
200
400
600
800
1000
STANDBY TIME, HRS.

usable amount. Standby with this method is the time period between filling the storage vessel, buildup to the maximum operating pressure, and venting of the fluid until the remaining usable quantity is identical to that contained in the shrouded and non-venting systems. The standby time is increased by increasing the number of discrete radiation shields and the vapor-cooling of one of them.

Because of the importance of system weight at the end of standby, both this value and the system weight at launch are considered in Figures 20 through 22. System weight at launch, for the shrouded systems, consists of the primary storage vessel, primary fluid, shroud vessel, shroud fluid, insulation, and outer shell weights. System weight at the end of standby for the shrouded systems is calculated on the assumption that the shroud fluid is completely vaporized, but that no primary fluid has been expelled. For the conventional non-venting system, the weight at launch and at the end of standby are identical. System weight in this technique consists of the storage vessel, fluid, insulation, and outer shell weights. At the end of standby, the fluid weight considered is that amount required for use.

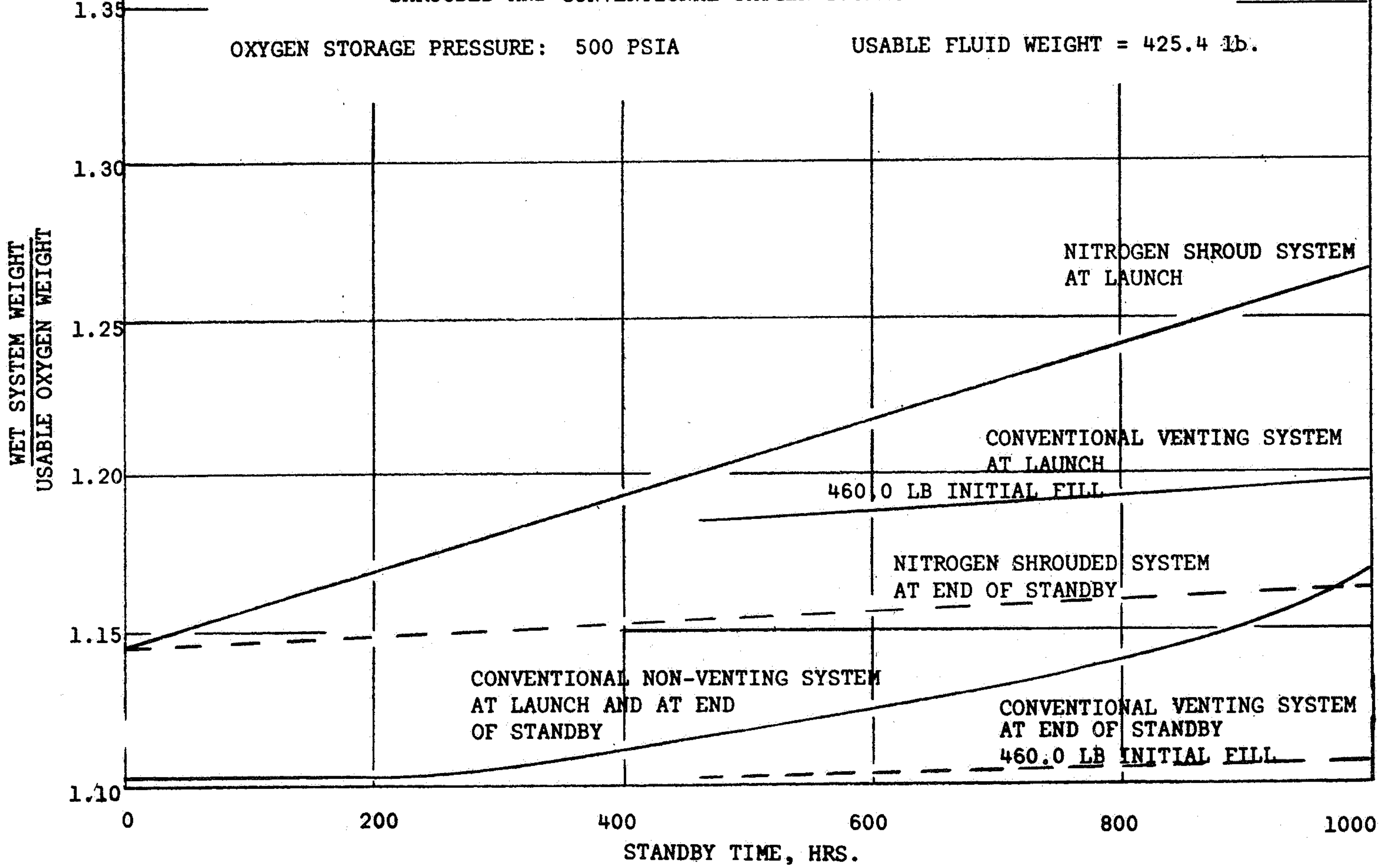
Figure 20 compares vented, non-vented, and shrouded systems for the storage of oxygen at 500 psia. This pressure was selected since the previous exercises with nitrogen cooling of oxygen showed that of the three pressures examined (3000, 1500, and 500 psia), the 500 psia system proved to be optimum for both weight and standby time considerations. The two-shield vapor-cooled nitrogen shrouded system examined in Figure 17 is used in this figure for comparison with the conventional storage systems. A common usable oxygen weight of 425.4 lb. is employed in all systems. The conventional vented storage system initially contains 460.0 lb. of oxygen and vents 34.6 lb. during standby. Initial oxygen loading conditions for the nitrogen shrouded system are 500 psia and 139.3°R. The conventional vented and non-vented systems are initially filled with oxygen at 14.7 psia and 162.3°R.

Throughout the standby time period examined, the vented oxygen storage system is lighter than the nitrogen shrouded system both in terms of weight at launch and weight at end of standby. The two plots for the vented system as shown originate at 460 hours standby. These plots are meaningless for smaller standby times, since the 460 hour weight ratios represent an unshielded unit. The weight at launch for the nitrogen shrouded unit is nearly the same as for the vented system in the 460-500 hour standby period, but the difference increases with longer standby. Weight at the end of standby for the shrouded system is always considerably higher than that of the vented system. The non-vented oxygen storage system is superior to both the vented and the nitrogen shrouded systems in at-launch weight, for the standby period examined. For all standby times, the specific vented system used is lighter in end-of-standby weight than the non-vented system. At 950 hours, the nitrogen shrouded unit is shown to be lighter than the non-vented system at end of standby.

For standby times in excess of 1000 hours, the trends shown in Figure 20 indicate that the vented system will eventually become lighter than the non-vented system in at-launch weight. However, the vented system chosen is not necessarily the optimum for the given usable fluid weight. It is conceivable that a vented system could be determined that would remain lighter, at-launch, than the non-vented unit. At the same time,

FIGURE 20

COMPARISON OF EFFECTIVE FLIGHT SYSTEM WEIGHT AND STANDBY TIME
FOR
SHROUDED AND CONVENTIONAL OXYGEN STORAGE SYSTEMS



additional insulation techniques applied to the nitrogen shrouded system would result in lower system weight for a given standby time, both at launch and at end of standby. The shrouded system therefore could be lighter than both the vented and non-vented systems at launch. At end of standby, it does not appear that the shrouded system would be lighter than the vented system.

Shrouded, non-vented, and vented systems for the storage of hydrogen at 500 psia are compared in Figure 21. The neon-shrouded system is identical to the two-shield vapor-cooled system examined in Figure 18. All systems consider a usable hydrogen weight of 24.0 lb. The vented storage system initially contains 28.0 lb. of hydrogen, and vents 4.0 lb. to result in a usable fluid weight comparable to the other systems. Initial hydrogen loading conditions for the neon shrouded system are 500 psia and 48.8°R. The vented and non-vented systems are initially filled with hydrogen at 14.7 psia and 36.7°R.

For standby times in excess of 170 hours, the neon shrouded system is lighter than a non-venting system at the end of standby. The neon shrouded system is also lighter than the non-venting system at launch, if required standby is greater than 190 hours.

The specific venting system considered appears to be superior to the neon shrouded system at launch, for the standby times examined (up to 1000 hours). The shrouded system does display an end-of-standby weight advantage after 900 hours standby. It was beyond the scope of this report to determine an optimum venting system for the specific usable fluid quantity of 24.0 lb. hydrogen. It is therefore possible that a venting system could be designed that would be superior to the shrouded system from an end-of-standby weight consideration. The trends of the curves in Figure 21 show, however, that for an extremely long standby beyond 1000 hours, the neon shrouded hydrogen system with two shields (one vapor-cooled) is the weight optimum system. Further insulation of the shrouded system with more than two shields would reduce the system weight ratio for a given standby time. This was not considered in this report, but the results of such an examination could show definite weight advantages for the neon shrouded system over a conventional venting system.

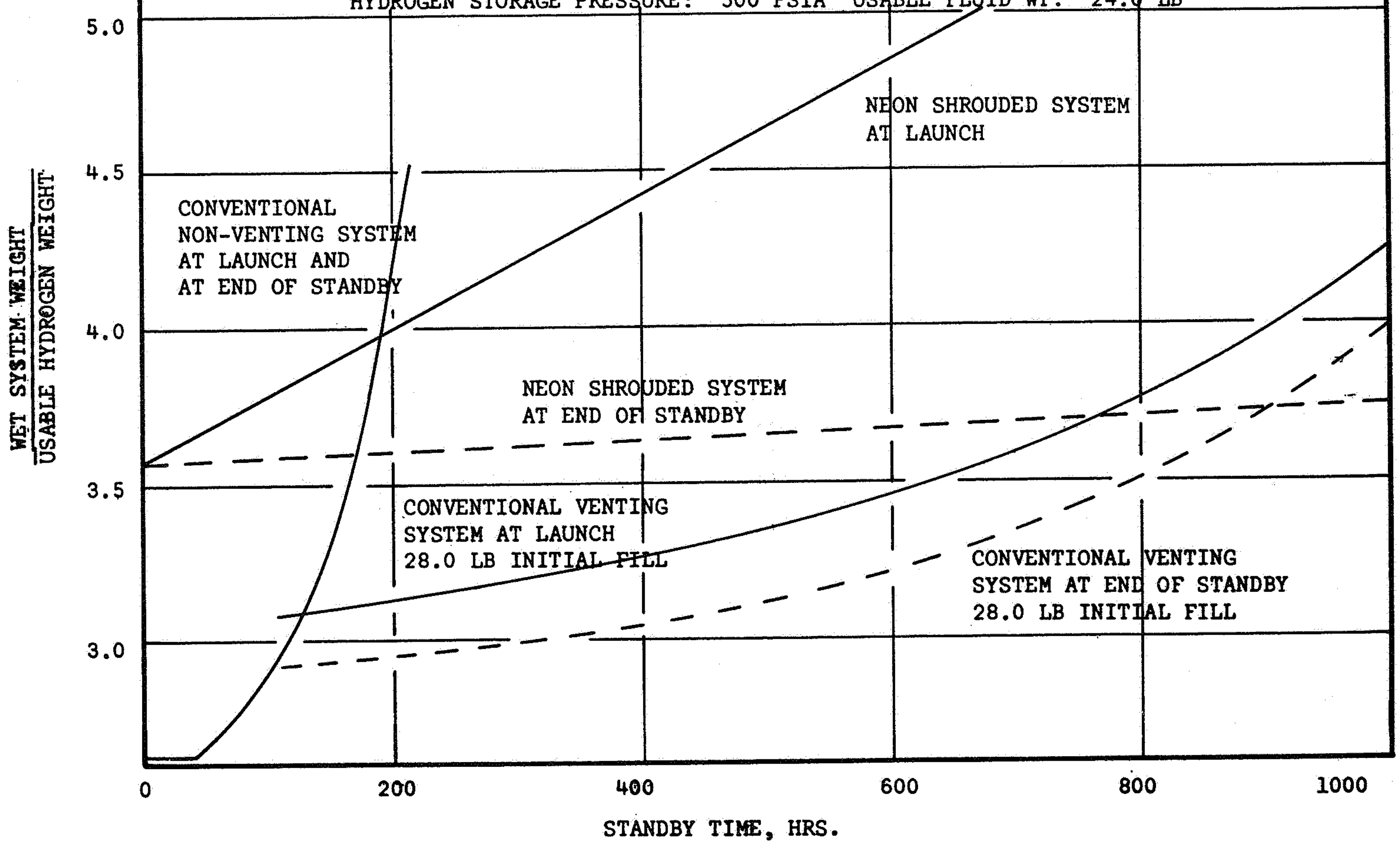
Figure 22 compares the storage of helium at 1500 psia in shrouded, vented, and non-vented systems. The shrouded system shown is the same two-shield vapor-cooled in-flight cooling system examined in Figure 19. A common usable helium weight of 52.7 lb. is employed in all systems. The vented storage system initially contains 60.0 lb. of helium and vents 7.3 lb. during standby. Initial helium loading conditions for the hydrogen shrouded system are 1500 psia and 36.7°R. The vented and non-vented systems are initially filled with helium at 200 psia and 10°R.

For standby times in excess of 330 hours, the hydrogen shrouded system is shown to be lighter than the non-venting system at the end of standby. At launch, the hydrogen shrouded system weight is less than that of the non-venting system for standby times of 345 hours or longer.

FIGURE 21

COMPARISON OF EFFECTIVE FLIGHT SYSTEM WEIGHT AND STANDBY TIME
FOR
SHROUDED AND CONVENTIONAL HYDROGEN STORAGE SYSTEMS

HYDROGEN STORAGE PRESSURE: 500 PSIA USABLE FLUID WT: 24.0 LB



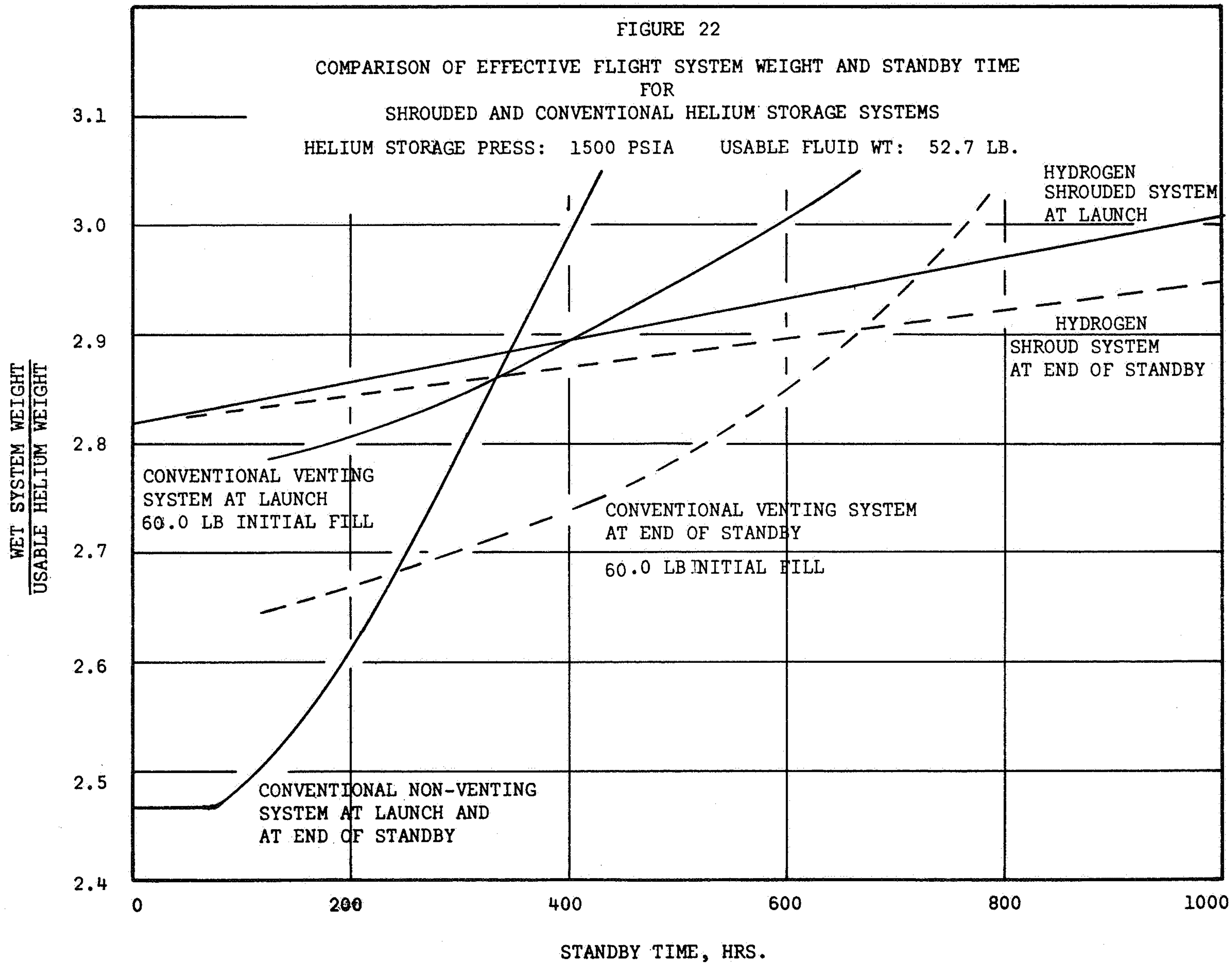


FIGURE 22

COMPARISON OF EFFECTIVE FLIGHT SYSTEM WEIGHT AND STANDBY TIME
FOR
SHROUDED AND CONVENTIONAL HELIUM STORAGE SYSTEMS

HELIUM STORAGE PRESS: 1500 PSIA USABLE FLUID WT: 52.7 LB.

The 60.0 lb. venting system is shown to be superior to the hydrogen shrouded system in at-launch weight until standby exceeds 400 hours. After 620 hours standby, the hydrogen shrouded system proves to be lightest in end-of-standby weight. Further examination of venting systems having a usable helium weight of 52.7 lb. could result in an optimum venting storage system, with a lighter system weight than the hydrogen shrouded system, for all standby times shown in Figure 22. Conversely, additional insulating techniques (more vapor-cooled radiation shields) could be applied to the shrouded system to increase standby for a given system weight. It was not possible to examine all such alternatives in this report.

One of the most significant advantages realized with a hydrogen-cooled helium system is related to the transfer procedures required to initially fill the primary storage vessel. Due to the low heat of vaporization, low critical pressure, and low specific heat of helium, it is extremely difficult to transfer liquid helium to a cryogenic storage system. For this reason, cryogenic helium is currently stored at supercritical pressures in ground storage dewars and then transferred to the flight storage system. Typical storage and transfer conditions are similar to that used in this examination, i.e., 200 psia and 10°R. Transfer of helium at this state is still extremely difficult, again due to the low specific heat. The initial flow of helium to a flight storage system results in large helium losses as the storage vessel cools from some initial pre-cooled temperature to the final storage temperature. Transfer lines must be well insulated to result in minimal heat transfer to the helium upstream of the flight storage vessel. Although extreme insulation techniques are employed throughout the transfer system, the cool-down and loading times are undesirable, helium losses are high, and storage densities are not optimum. The hydrogen shrouded helium storage system developed under this contract represents an improved method for the storage of helium at high densities, with minimized loading problems.

Ground support equipment for storing helium will be simplified with the hydrogen shrouded helium storage technique, since the helium need only be stored at ambient temperature and high pressure. The primary storage vessel will be pre-cooled to the final storage temperature of the helium by loading the shroud with liquid hydrogen. With the primary storage vessel closed, gaseous helium is transferred at high pressure into the unit. The filling operation is complete when the vessel contains helium entirely at liquid hydrogen temperature and the helium transfer pressure, corresponding to some high helium density. Since hydrogen will be vaporizing during the fill process, a continuous supply must be transferred to the shroud until the loading is complete.

III SYSTEM DESCRIPTION AND OPERATION

The Cryogenic Shroud System developed under this contract is shown in Figure 23. The system was designed solely for testing the shroud concept, and therefore certain components are not weight-optimized. Manual valves are located on the tank mount carriage to provide for filling, venting, and pressurization of both the inner and shroud vessels. Pressure relief valves are provided for both vessels. The inner vessel operating pressure is automatically maintained by actuation of a pressure switch which controls an internal heater and motor-fan.

DEWAR DESCRIPTION

The Shroud Tank assembly is shown in Figure 24. Physical Characteristics of the tank are presented in Table VII.

Figure 24 shows that the tank assembly consists of an inner pressure vessel which is surrounded by and permanently attached to an integral shroud vessel. The shroud vessel contacts the outer surface of the inner vessel at six specific locations where the shroud shell is formed into inverted cups to accommodate the radial support bumpers. The shroud vessel is also heli-arc welded to the Inconel 718 inner vessel fill and vent fittings, resulting in a permanent shroud-inner assembly.

Functioning units within the inner vessel are a motor-fan assembly, calrod heater, capacitance-type quantity sensor, and a single copper-constantan thermocouple. The motor-fan is identical to that utilized in Apollo hydrogen tankage. Its purpose is to eliminate temperature stratification in the inner vessel fluid by perodic stirring of the fluid. This particular motor-fan unit was inoperative, a point that will be discussed later in more detail.

The calrod heater, located in the lower portion of the inner vessel, is wrapped lengthwise on the support tube containing the motor-fan unit. During periods of high supply flow from the inner vessel, or for rapid pressurization of the fluid contents from some low pressure state, the 171.5 watt heater is actuated to maintain operating pressure. An interleaf parallel plate linear capacitance type probe is mounted in the upper portion of the inner vessel, electrically insulated from its support tube. The element is not designed for liquid level measurement at two-phase, subcritical pressure. At supercritical pressure (single phase) a density change within the container is accompanied by a change in the total capacitance of the probe, due to the variation of the dielectric constant of the stored single-phase fluid with its density. The fan and heater operation are simultaneously controlled by a pressure switch (to be discussed later), which senses inner vessel pressure and energizes the fan and heater when pressure degrades below the operating pressure of 1000 psig.

The fill and vent tubes are heli-arc welded to the inner-vessel fittings, and provide the means of transferring fluid to and from the inner vessel. The fill tube is located at the extreme bottom point of the vessel, and the vent tube

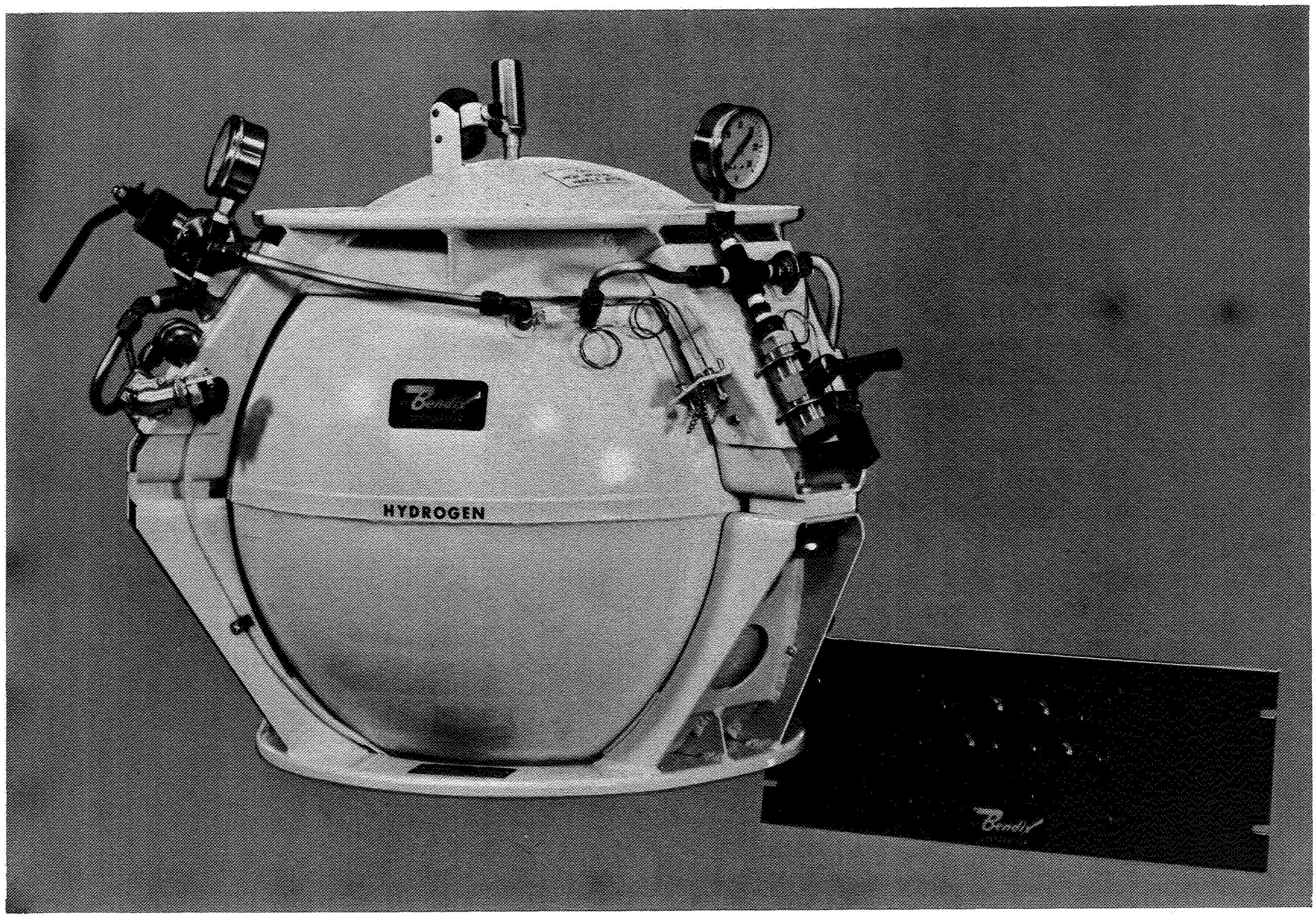
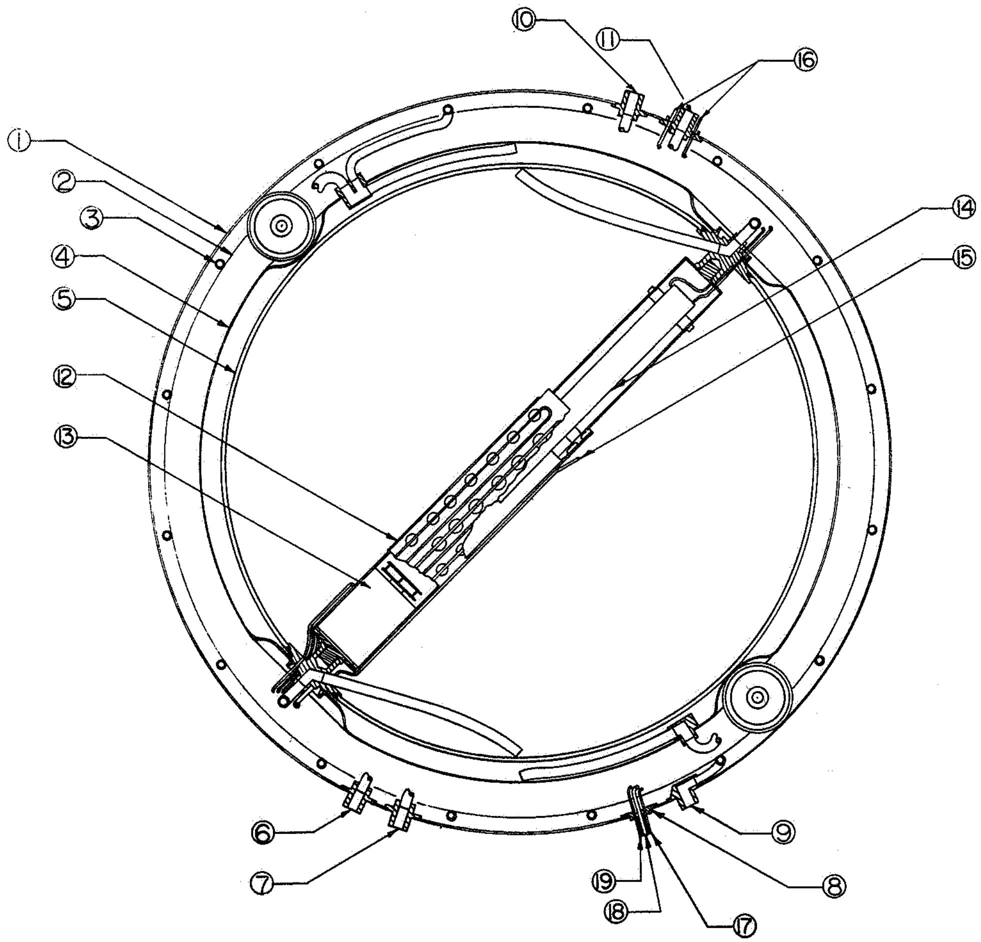


Figure 23

CRYOGENIC SHROUD SYSTEM



1. OUTER SHELL
2. RADIATION SHIELD
3. VAPOR-COOLING TUBE
4. SHROUD VESSEL
5. INNER VESSEL
6. SHROUD FILL FITTING
7. INNER FILL FITTING
8. ELECTRICAL LEAD FITTING
9. SHROUD VAPOR-COOLING VENT FITTING
10. SHROUD VENT FITTING
11. INNER VENT FITTING
12. CALROD HEATER
13. MOTOR-FAN
14. CAPACITANCE PROBE
15. THERMOCOUPLE
16. CAPACITANCE PROBE LEADS
17. THERMOCOUPLE LEAD
18. CALROD HEATER LEAD
19. MOTOR FAN LEADS (4)

Figure 24

ASSEMBLY - SHROUD TANK

TABLE VII

SHROUD TANK PHYSICAL CHARACTERISTICS

Dry Tank Weight	76.6 lb.
Tank Components	
Inner Vessel	
Material	Inconel 718 (annealed)
Inner Diameter	17.440 in.
Wall Thickness	0.145 in.
Usable Volume	1.600 ft^3
Operating Pressure	1000 psig
Proof Pressure	1400 psig
Shroud Vessel	
Material	Inconel 718 (annealed)
Inner Diameter	18.990 in.
Wall	0.030 in.
Usable Volume	0.382 ft^3
Operating Pressure	14.7 psia
Proof Pressure	40 psig
Vapor-Cooled Discrete Shield	
Material	Aluminum 6061-0
Inner Diameter	21.090 in.
Wall	0.020 in.
Outer Shell	
Material	Stainless Steel 304L
Inner Diameter	22.000 in.
Wall	0.030 in.
Inner Vessel Fill and Vent Tubing	
Material	Stainless Steel 304L
Outer Diameter	0.3125 in.
Wall Thickness	0.022 in.
Shroud Vessel Fill and Vent Tubing	
Material	Stainless Steel 304L
Outer Diameter	0.3125 in.
Wall Thickness	0.022 in.
Vapor-Cooling Tubing	
Material	Copper (annealed)
Outer Diameter	0.250 in.
Wall Thickness	0.030 in.
Total Length	31 ft.
Internal Components	
Quantity Sensor	
Type	Interleaf Parallel Plate Capacitance
Output in Medium (Submerged)	
Air 70°F	409 Pico Farads
Liquid Oxygen	590 Pico Farads
Liquid Nitrogen	580 Pico Farads
Liquid Hydrogen	493 Pico Farads
Temperature Sensor	
Manufacturer	American Standard
Type	Copper-Constantan Thermocouple

TABLE VII (Cont.)

SHROUD TANK PHYSICAL CHARACTERISTICS

Motor-Fan	
Manufacturer	Globe
Type	Cryogenic Destratification-Hydrogen
Input	200 VAC, 3 Phase, 900 cps
Operating Power	5.0 Watts Maximum
Heater	
Manufacturer	American Standard
Type	Resistance-Nichrome
Input	115 VAC, Single Phase, 60 cps
Power	171.5 Watts

is placed at the uppermost point. If the inner vessel is initially charged with liquid cryogen, this location of the upper vent tube permits essentially 100% fill.

Capacitance probe electrical leads exit from the inner vessel through silver-brazed joints in the inner vent fitting. In the same manner, the four motor-fan leads, the thermocouple lead, and the heater lead, exit through the inner fill fitting. The fill and vent transfer lines are silver-brazed into their respective inner vessel fittings, and these tubes subsequently pass through the radial support bumpers in the vacuum annulus. The electrical leads exiting from the inner fill fitting pass through one radial bumper as a single unit, while the two capacitance probe leads accompany the inner vent tube through two of the three upper radial support bumpers.

Although there is a weldment of the shroud vessel to the two inner vessel fittings, the shroud and inner vessels themselves are completely separate and form two individual storage vessels in one integral assembly. Thus, the shroud vessel is equipped with its own fill and vent fittings. The fill and vent tubes are silver-brazed in these fittings and permit bottom introduction of the fluid to the vessel and a 100% fill capability.

The shroud fill transfer tube passes through one of the three lower radial support bumpers in the vacuum annulus. Two transfer tubes exit from the shroud vent fitting. One passes through a single upper radial bumper, while the other is joined to the vapor-cooling line which is attached to the single discrete radiation shield in the vacuum annulus. The capability of transferring shroud vent gas through the normal vent line or through the vapor-cooling network will be discussed later.

The shroud vessel-inner vessel assembly is supported within the outer shell (vacuum jacket) by six Kel-F radial support bumpers. The low thermal conductivity of this type of vessel support and the small contact area at support locations results in very low conductive heat transfer from the environment to the fluid contents. A single discrete radiation shield, located in the vacuum annulus between the shroud-inner vessel assembly and the outer shell, reduces radiation heat transfer to the fluid contents. The advantages and effects of this type of insulation have been discussed earlier in this report. The shield is isothermally-mounted on the fill and vent transfer tubing and the electrical lead bundle at six locations by Kel-F spacers. Cut-outs in the shield for bumper and tubular emergence are minimized to prevent black-body radiation heat transfer. The radiation shield provides essentially 100% coverage of the shroud-inner vessel unit. Copper tubing is soldered along its entire length on the aluminum shield's outer surface. This provides continuous contact of the vapor-cooling tubing network with the shield, which enables efficient heat exchange between the shroud fluid and the shield. The low temperature environment thus surrounding the shroud vessel-inner vessel assembly when vapor-cooling is used results in further reduction of radiant heat transfer, as described previously.

The fill and vent lines, as well as the electrical leads, exit from the dewar through silver-brazed joints in the stainless steel 304L fittings. These fittings are heli-arc welded in the 304L outer shell. The vacuum annulus is evacuated to a high vacuum of 10^{-7} mmHg or more to eliminate convective heat transfer to the fluid contents. Two quantities of chabazite getter are attached to the assembly at the two inner vessel fittings, and within the vacuum annulus. The getter assists in maintaining a high, stable vacuum. The outer surface of the shroud vessel and the entire radiation shield are silver-plated to result in low-emissivity surfaces during system operation. The inner surface of the outer shell, being essentially at environmental temperature, is copper-plated to provide a low-emissivity surface at this relatively higher temperature. The conductive heat paths in the transfer tubing and electrical-lead bundle are minimized by the long length of these paths. In addition, the tubing and leads are not silver plated, to avoid the higher thermal conductivity of silver as opposed to the stainless steel from which the tubes and lead sheaths are fabricated.

SYSTEM DESCRIPTION

Physical characteristics of the Cryogenic Shroud System are presented in Table VIII.

The shroud tank assembly is supported within a mount carriage at the points of contact between the radial bumpers and outer shell so that all vessel loads are carried directly from the bumpers through the outer shell and into the mount carriage. The tank-mount carriage assembly (without external plumbing) is shown in Figure 25. The system is supported at three vibration-damper locations.

The complete Cryogenic Shroud System shown in Figure 23 is represented schematically in Figure 26. Manually operated fill and vent ball valves for the shroud vessel and the inner pressure vessel are mounted on the mount carriage. Two vent valves for the shroud vessel permit either normal shroud venting directly to the atmosphere or vapor-cooling of the discrete radiation shield. Pressure relief valves are provided for both the shroud vessel and the inner vessel in the event excessive pressurization of either cryogen occurs.

The motor-fan and heater operation are controlled by a pressure switch, located on the mount carriage. This pressure switch senses inner vessel pressure and energizes the fan and heater when pressure degrades below the operating pressure of 1000 psig. Power for operation of the fan and heater is controlled by the control panel which is a part of the system. Manual operation of the fan at any time can be accomplished (as stated previously, the motor-fan in the contract system is currently inoperative). The heater operation is controlled only by the pressure switch, with no manual override.

An ion pump attached to the evacuation tube of the tank (not shown in Figure 26) serves two purposes. It is used to remove out-gassed materials from the vacuum annulus, and to provide a means of determining the pressure within the vacuum annulus. The ion pump operates from a separate power supply provided with the system.

A single electrical connector mounted on the mount carriage provides for power input to the motor-fan and heater through the control panel. Individual outputs are provided for monitoring the thermocouple and capacitance probe outputs.

TABLE VIII

CRYOGENIC SHROUD SYSTEM PHYSICAL CHARACTERISTICS

Dry Tank Weight (from Table VII)	76.6 lb.
External Components Weight	32.7 lb.
Total Dry System Weight	109.3 lb.
External Components	
Inner Fill and Inner Vent Valves	
Manufacturer	Hydromatics
Type	Manual Ball Valve
Orifice	0.375 in.
Seats	Kel-F
Shroud Fill and Shroud Vent Valves	
Manufacturer	Hoke
Type	Manual Ball Valve
Orifice	0.375 in.
Seats	Teflon
Shroud Vapor-Cool Vent Valve	
Manufacturer	Hoke
Type	Manual Ball Valve
Orifice	0.250 in.
Seats	Kel-F
Inner Pressure Relief Valve	
Manufacturer	James, Pond & Clark
Cracking Pressure	1050 psig
Maximum Flow	180 lpm @ 1320 psig
Shroud Pressure Relief Valve	
Manufacturer	Bendix
Cracking Pressure	25 psig
Pressure Switch	
Manufacturer	International Controls Corp.
Switch Opening Pressure (on inc. press.)	1000 psig
Switch Closing Pressure (on dec. press.)	950 psig
Power Rating	5 amp., 115 VAC
Ion Pump	
Manufacturer	Ultek
Capacity	0.2 ltr/sec. 10^{-5} to 10^{-8} torr.
Mount Carriage	
Material	Alum. 6061-T6
Inner Vessel and Shroud Vessel	
Fill and Vent Tubing	
Material	Alum.
Outer Diameter	0.375 in.
Wall Thickness	0.030 in.
Vibration Dampers	
Manufacturer	Robinson

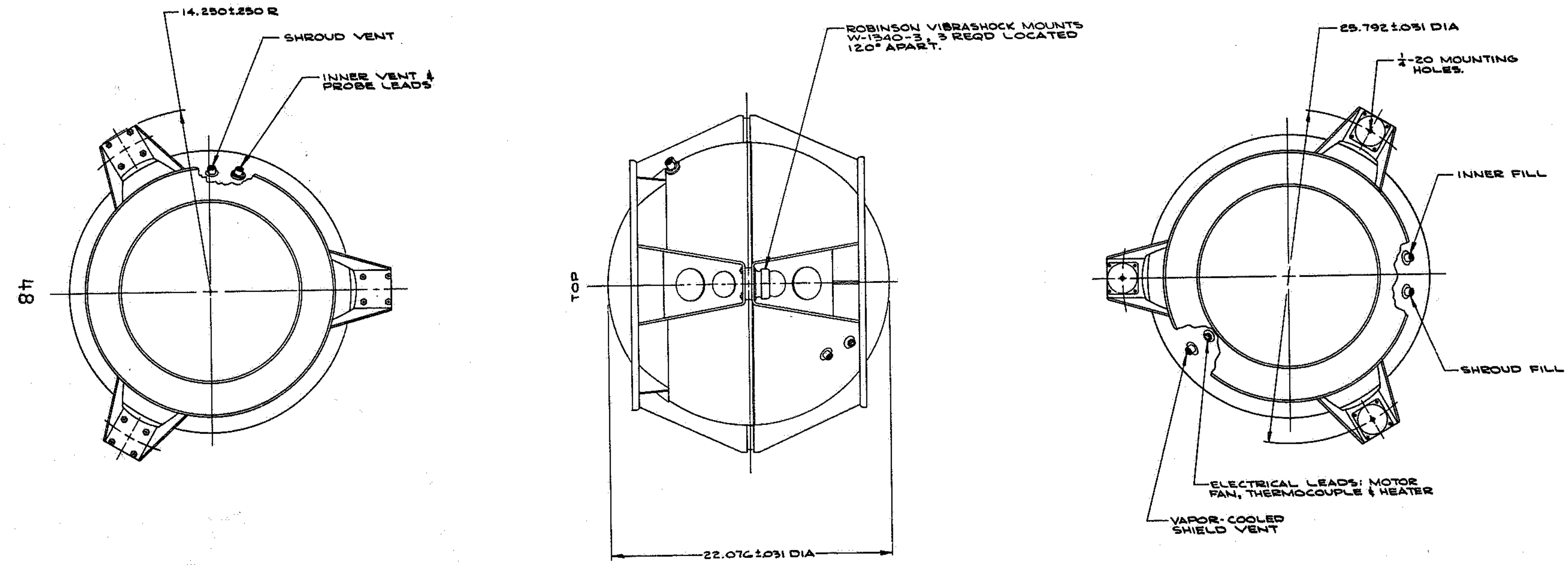


Figure 25

ASSEMBLY - SHROUD TANK MOUNT CARRIAGE

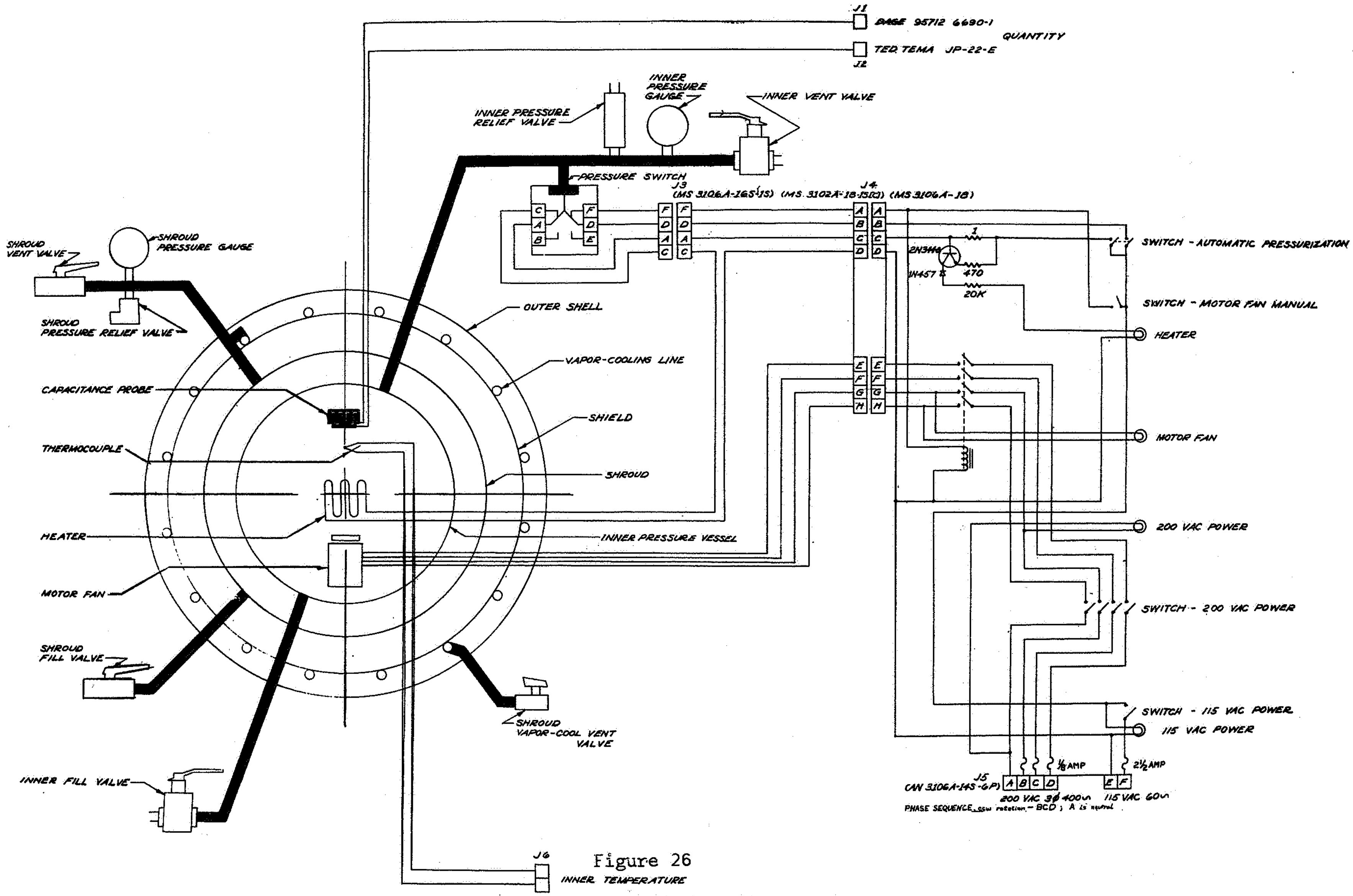


Figure 26

SCHEMATIC - CRYOGENIC SHROUD SYSTEM

SYSTEM OPERATION

The normal operation of the Cryogenic Shroud System would involve pre-cooling of the inner vessel by filling the shroud with the desired secondary refrigerant. To avoid cryopumping of atmospheric gas into the inner vessel as it is cooled, the vessel should be pressurized and sealed with a gas having a lower boiling point than the shroud liquid. Preferably, the gas should be the same as the primary fluid to be used in the inner vessel. The shroud liquid source should be pressurized to 15-30 psig to insure a continuous flow of liquid into the shroud. During the fill process, both the shroud vent valve and the shroud vapor-cool vent valve should be open to allow rapid expulsion of the vaporized shroud fluid when it initially contacts the warm shroud and inner vessel walls. When the container is sufficiently cooled to the shroud liquid temperature, the shroud vessel will fill with liquid. Completion of fill is determined when liquid issues from the shroud vent valve. Continued transfer to the unit will eventually result in liquid issuing from the shroud vapor-cool vent valve.

During the process of filling the inner vessel, it is desirable to maintain a low flow of secondary refrigerant into the shroud. This is recommended since the primary fluid transfer, in either its liquid or gas phase, results in heat input to the system and therefore vaporization of the shroud liquid. When transferring liquid into the inner vessel (eg., a liquid nitrogen-liquid oxygen vented test), the inner vessel vent valve is opened, and the 15-30 psig pressurized liquid source is transferred through the inner fill valve. For many of the combinations of primary and secondary fluids discussed earlier in this report, the primary fluid will be transferred in a gaseous state into the inner vessel at a pre-selected storage pressure. For example, when using liquid hydrogen in the shroud as the secondary refrigerant, and helium as the primary fluid, the inner vessel would be closed at the vent valve and gaseous helium would be transferred at 1000 psig into the vessel through the inner fill valve. With a well-regulated source of helium transferred at pressure in such a manner that the inner vessel relief valve does not open, the inner fill valve may be left open until flow ceases. If liquid hydrogen has been maintained in the shroud, and if temperature stratification has been avoided within the inner vessel, the vessel will now be filled with helium at liquid hydrogen temperature and at 1000 psig. The inner vessel fill valve is then closed and the helium transfer line removed.

Maximum shroud cooling effects will be obtained when the shroud vent valve is closed and the shroud vapor-cool valve is left open. This will permit a transfer of the vented shroud fluid through the vapor-cooling tubing attached to the radiation shield.

The system control panel and pressure switch provide an automatic pressure control of the inner vessel fluid contents. Attachment of the appropriate mating connectors should be made from the control panel to the connector on the shroud system (J4), and to 200 VAC, 3-phase, 400 cps and 115 VAC, single-phase, 60 cps power supplies (J5). When the pressure is below 1000 psig, and in the process of being increased, the pressure switch is automatically closed and the heater and motor fan are energized. At 1000 psig the pressure switch opens and the power circuits to the heater and motor fan are likewise opened. If pressure then declines, the pressure switch will again close and energize the fan and heater. The motor-fan operation is desired during operation of the heater to result in a flow of the fluid to and over the heating circuit. A manual overide switch is provided to permit motor-fan energization at any time to allow for elimination of fluid temperature stratification. The heater is energized only when the pressure switch is closed, i.e., at pressures below 1000 psig. Pressure relief at 1050 psig is accomplished by the inner pressure relief valve. A pressure relief valve, set at 25 psig, is provided for pressure control in the shroud vessel.

Single-phase quantity measurement for the primary fluid is accomplished by determining capacitance output from the interleaf capacitance probe. The capacitance range for 100% full to empty depends upon the primary fluid used and its temperature and pressure. Capacitance output for the specific probe utilized in this contract unit is 409 picofarads in air at 70°F and 14.7 psia. When submerged in liquid cryogens at 14.7 psia, the capacitance outputs are 590, 580 and 493 picofarads for oxygen, nitrogen, and hydrogen, respectively. Increasing the fluid density, by increasing the fluid pressure and/or decreasing fluid temperature, results in an increasing capacitance output from the probe. Output is linear throughout the entire range of full to empty due to the parallel plate construction.

The temperature of the primary fluid in the center of the inner vessel is determined by voltage measurement of the copper-constantan thermocouple output. An electrical connection for potentiometer usage is provided on the system mount carriage (J6).

IV PROGRAM DESCRIPTION

Three of the basic objectives of this program were as follows:

1. To demonstrate that the integrally-mounted shroud design can be fabricated.

2. To demonstrate the concept of shroud cooling in a flight-type cryogenic storage system.

3. To demonstrate the concept of using vapor-cooled, discrete radiation shield insulation for flight-type cryogenic storage systems.

FABRICATION

The primary storage vessel was fabricated from Inconel 718 rather than titanium for several reasons. Not only is the metal more economical, but there is more knowledge and data available concerning fabrication techniques. Since it is compatible with oxygen, oxygen may be used as the primary fluid if so desired. It also has adequate ductility over the cryogenic temperature ranges, and it can be readily formed by the hydroform process.

To prove that the hydroformed and annealed Inconel 718 hemispheres were adequate to withstand the 1000 psig inner vessel operating pressure after girth welding, and to also prove that the completed inner vessel need not be age-hardened, a hydrostatic proof pressure test was performed on a prototype inner vessel. Following pressurization to the proof pressure of 1500 psig, vessel measurements indicated that the annealed condition of the material was adequate for the design operating pressure.

Inconel 718 was also used for the shroud vessel material for compatible heli-arc welding to the inner vessel fittings (also Inconel 718), to insure a vacuum and pressure tight shroud. The shroud hemispheres were formed by the hydroform process.

Aluminum 6061 was selected as the shield hemisphere material because of its light weight and ease of forming by the hydroform process.

The outer shell was fabricated from 304L stainless steel because it is both economical and adaptable to forming, and especially because the material does not offer a serious weight penalty. The entire system was not designed to be a flight-weight unit, as evidenced by the thick-walled, unaged Inconel 718 inner vessel. It was not necessary, therefore, to consider aluminum for the outer shell material, and aluminum to stainless steel joints were then not required for the outer fill and vent fittings.

The inner vessel and shroud vessel fittings were fabricated from Inconel 718 for compatible welding to the shells. A problem area developed when the stainless steel-sheathed motor fan, thermocouple, and heater leads were silver brazed into the inner vessel fill fitting, prior to welding this fitting in its inner hemisphere. During the brazing process a crack developed in the fitting in the area of the feed-through holes, The problem was determined to be stress corrosion cracking due to the silver brazing process. A new fitting was fabricated dimensionally identical to the original fitting. The fitting was then solution annealed to remove machining stresses, and thus avoid the stress corrosion problem. No problems were encountered when the electrical leads were satisfactorily brazed into this fitting.

All weld and silver-braze joints in the tank relating to vacuum integrity were extensively helium mass spectrometer leak tested to prove zero leakage at 10^{-8} atm-cc/sec.

Following silver plating of the shroud-inner vessel assembly and the vapor-cooled shield and copper plating of the inner surfaces of the outer shell hemispheres, the tank was assembled within an inert atmosphere enclosure to prevent tarnishing of plated surfaces. The system was extensively leak checked following weldment of the outer shell girth and external fittings. An evacuation-bakeout period was initiated, with a bakeout temperature of 300°F maintained throughout the process. Bakeout was limited to this temperature for two reasons: (1) the radial support bumpers were fabricated from Kel-F, which is superior to Teflon in terms of minimum thermal conductivity, but is limited to 350°F for strength retention, and (2) the presence of the motor-fan unit necessitates avoidance of high bakeout temperatures.

The evacuation-bakeout period required a total of 19 days, with a minimum pressure of 2.6×10^{-7} mmHg attained immediately prior to the sealing of the evacuation tube. Initial cool-down of the inner vessel and shroud was performed with liquid nitrogen. The loss rates indicated that the pressure within the vacuum annulus had increased, due either to an improper cold weld in the evacuation tube pinch-off or to outgassing from some material within the vacuum annulus. The unit was immediately re-evacuated and baked out at the same temperature for 6 days, following which it was pinched off and again cooled with liquid nitrogen. Heat input calculations based upon the total nitrogen weight loss indicated that the vacuum was stable, with the loss rate determined to be 7.9 Btu/hr.

A problem arose when the shroud vessel was initially filled with liquid hydrogen, as partial loss of vacuum resulted. It was determined through helium leak testing that the problem area was near the inner vessel-shroud weld attachment at the inner vessel fill fitting. The area was repaired by cutting off the outer shell and welding a small crack which had developed in the shroud shell near the inner vessel fitting. Following extensive helium leak testing, and pressure and temperature cycling to prove the repaired area, the shroud vessel and shield hemispheres were cleaned and silver-plating was accomplished. New outer shells were modified, cleaned, and copper-plated. The unit was reassembled in the inert-gas booth and placed on the evacuation pumps for the bakeout process. An 18 day evacuation bakeout period was required, with a vacuum of 7.5×10^{-8} mm Hg attained at pinch-off. The total liquid nitrogen vented loss rate from the shroud and inner vessel was determined to be 8 Btu/hr, which was practically the same value obtained prior to the turn-around assembly process. The unit was later filled with liquid hdyrogen, with no problems resulting.

The total turn-around time, from discovery of the problem when the unit was filled with liquid hyrogen until the reassembled unit was again filled with hydrogen, was only 48 days. This time included disassembly, repair, replating, reassembly, and evacuation bakeout. In addition, repeatability of insulation quality was proven by comparable heat leak values.

Energization of the motor-fan unit following liquid hydrogen testing indicated that the motor-fan was inoperative. A detailed checkout showed that the problem was not electrical, in that current and voltage relationships corresponded to those expected for a stalled unit. Oscilliscope readings did not show a back EMF developed when power to the motor was turned off, thus indicating that the fan shaft was not rotating. Since the motor-fan is within the inner vessel and therefore inaccessible, no repair was possible. Various methods were employed unsuccessfully to free the locked rotor shaft, including temperature shock and cleaning. It is assumed that some foreign matter became lodged in the motor air-gap, causing the blockage.

TESTING

The testing program performed on the system at Bendix consisted of (1) liquid nitrogen venting tests, vapor-cooled and non vapor-cooled, (2) liquid hydrogen venting tests, vapor-cooled and non vapor-cooled, and (3) pressure build-up in the inner vessel with liquid hydrogen in both vessels. The test program was limited because of delivery scheduling. A more extensive program was performed by the Thermochemical Test Branch, Propulsion and Power Division, NASA Manned Spacecraft Center, Houston, Texas at their Thermochemical Test Facility.

The results of the vented tests performed on the system at Bendix are presented in Table IX. The loss rate from the inner vessel for the liquid nitrogen system vented test was determined by measuring the flow with a wet-test meter. The shroud flow rate was then determined from the total system weight loss, measured with a recording scale. For liquid nitrogen vented tests with liquid only in the shroud, the flow rates were determined from wet-test meter measurements. The total system flow rates for the liquid hydrogen system vented tests were obtained from recording scale measurements.

The effect of vapor-cooling is shown by the results of the liquid hydrogen system venting tests. With direct venting of the shroud fluid to atmosphere, the total system heat leak was 4.2 Btu/hr. Transfer of shroud vent fluid through the vapor cooled shield resulted in a total system heat leak of 3.0 Btu/hr. Ambient temperature was slightly lower during the vapor-cooled test, however the temperature difference was not great enough to be solely responsible for the decrease in heat leak.

A pressure build-up test in the inner vessel was performed with the shroud filled with liquid hydrogen and vented through the vapor-cooled shield. The inner vessel was loaded to 92% full with liquid hydrogen and sealed. The pressure profile for this test is presented in Figure 27.

TABLE IX

CRYOGENIC SHROUD TANK VENTED HEAT LEAK TESTS

INNER VESSEL		SHROUD		SHIELD	TEST PERIOD HR.	AMBIENT TEMP °F	AVE. HEAT LEAK, BTU/HR.		
FLUID	PRESS. PSIA	FLUID	PRESS. PSIA				INNER VESSEL	SHROUD	TOTAL SYSTEM
LN_2	14.7	LN_2	14.7	NON VAPOR-COOLED	24	65	3.05	4.95	8.0
-	-	LN_2	14.7	VAPOR-COOLED	15	78	-	6.89	-
-	-	LN_2	14.7	VAPOR-COOLED	10	70	-	6.07	-
LH_2	14.7	LH_2	14.7	NON VAPOR-COOLED	20	65	-	-	4.2
LH_2	14.7	LH_2	14.7	VAPOR-COOLED	40	58	-	-	3.0

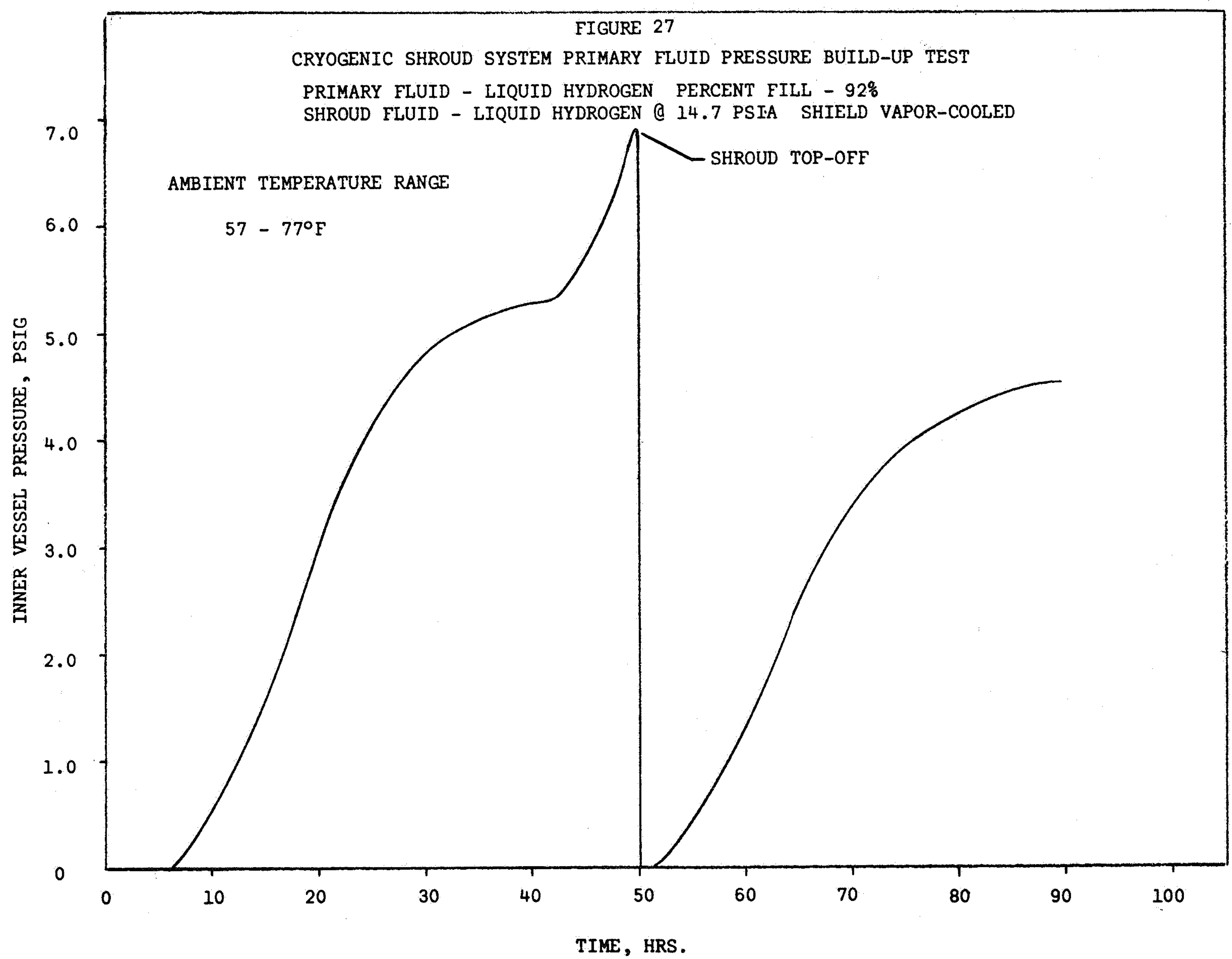


FIGURE 27

CRYOGENIC SHROUD SYSTEM PRIMARY FLUID PRESSURE BUILD-UP TEST

PRIMARY FLUID - LIQUID HYDROGEN PERCENT FILL - 92%
SHROUD FLUID - LIQUID HYDROGEN @ 14.7 PSIA SHIELD VAPOR-COOLED

The pressure increase during the test period was minimal. At 30 hours of test time, there was a decline in the pressurization rate and this was attributed to low ambient temperature. At 50 hours, the pressure had increased to 6.9 psig, with 44% of the original shroud fluid remaining. When the shroud was refilled with liquid hydrogen, there was a resulting drop in pressure to essentially zero psig. The pressurization rate following this drop was practically the same as during the earlier portion of the test.

The vented, vapor-cooled loss rate of hydrogen from the shroud during the first 50 hours of the test was determined from the recording scale measurements to be 3.63 Btu/hr. Based upon the initial fill of 92% and the inner vessel pressure rise observed during this period, the heat leak into the inner vessel was determined to be 0.64 Btu/hr. The total heat leak into the system was therefore 4.27 Btu/hr. The average ambient temperature during this period was 65°F.

The extended system standby for a cryogenic-shrouded system was demonstrated by the inner vessel pressure degradation obtained in this test when the shroud was topped off with liquid hydrogen. Continued shroud refilling of such a system would result in an extremely long ground standby period, with a very small pressure rise and no loss of primary fluid.

V CONCLUSIONS AND RECOMMENDATIONS

CONCLUSIONS

This program has successfully demonstrated the following:

1. An integrally-mounted cryogenic shroud system can be fabricated.

2. The cryogenic shroud concept is a feasible method for long duration storage of cryogenic fluids.

3. The vapor-cooled discrete radiation shield concept results in significant weight and thermal insulation advantages for cryogenic storage systems.

Examination of two types of liquid shroud designs -- the isothermally-mounted shroud and the integrally-mounted shroud -- resulted in the conclusion that the isothermally-mounted shroud system presents fabrication and assembly complications which are not offset by weight and thermal advantages, when compared with the integrally-mounted shroud design.

A study of various shroud-cooling applications showed the effects of primary fluid operating pressure and choice of secondary shroud fluid on both pre-launch and in-flight standby times. In particular, liquid hydrogen cooling of helium at 1500 psia storage pressure was shown to be the optimum system for helium storage in a shrouded unit (of the storage pressures examined). High density helium can be stored in such a system with minimization of ground support equipment capabilities.

From tests conducted on the Cryogenic Shroud System fabricated in this program, the feasibility of the concept was proven. Vapor cooling a discrete radiation shield with vented shroud fluid was shown to be a highly effective insulation technique for the contract system and for cryogenic storage systems in general. Extended ground standby capability of the system was demonstrated by the readily accessible method of refilling the shroud vessel.

RECOMMENDATIONS

Because of the successful results obtained from the Cryogenic Shroud System developed and fabricated in this program, it is recommended that the concept be applied to future space applications. Specifically, shroud cooling of helium by liquid hydrogen appears to be advantageous over present systems in terms of extended standby, system weight for long duration missions, and helium ground support equipment requirements. A weight-optimized, hydrogen-shrouded helium storage system should be fabricated based upon specific usable helium requirements and expected in-flight standby time for future lunar missions. The unit should employ the vapor-cooled discrete shield insulation concept which was also proven effective in this program.

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